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A TREATISE ON APPLIED HYDRAULICS

HYDRAULIC MEASUREMENTS

CENTRIFUGAL AND OTHER ROTODYNAMIC PUMPS

BY

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PREFACE

At a time when paper and typographical skill are alike hard to come by, it is proper that an author should be asked to make good his claim to have a book published at all. Some justification for this book may be found in the title : *Centrifugal and other Rotodynamic Pumps*. No other work has hitherto appeared with such a name, because there has been no single term in general use which would aptly describe the whole group of pumps which the book reviews. Such a term was badly needed, to designate not only centrifugal, screw, and propeller pumps, but also the entire range of machinery of a like nature, e.g. steam, gas, and hydraulic turbines, and fans and blowers. A common fundamental principle governs the working of all these machines. In every case, the interchange of energy between the fluid and the rotating blades depends upon the development of tangential acceleration in the fluid elements. The author's suggestion for this missing generic word was *Rotodynamic*. At the time when he communicated it to the technical press some years ago, engineers admittedly had many other things to think about ; but since nobody raised any objection to the word, the author proposes to use it until some other self-appointed philologist coins a better word.

Having gone to this amount of trouble to expose the kinship between radial-flow, mixed-flow, and axial-flow pumps, the author has naturally tried to confirm this close alliance throughout the book. Although as a matter of convenience separate chapters may be allocated to specific types of machine, the reader is often reminded that this is hardly more than an arbitrary distinction and that a continuous progression can be traced throughout the entire range of pumps. In other respects the sequence of chapters is based on the belief that " the best way to learn how to make a pump is to make one ". With a minimum of delay, the reader comes upon empirical rules for designing and testing pumps of various kinds, so that at least he shall have in mind an actual working machine before

beginning to analyse its performance. There may be minor drawbacks in dividing the material so sharply into sections devoted to Principles, Construction, Performance, and Installation : yet when one remembers the varying needs of users of technical books, the disadvantages may be accepted calmly. Probably few such users settle down to read the book steadily through from one end to the other ; they are more likely to consult such particular items as may concern them and accordingly they may prefer more or less self-contained sections, even if this involves some amount of repetition.

In regard to the treatment of rotodynamic pumps as such, the author has tried to develop to the fullest pitch the numerous valuable suggestions recently put forward by various writers for the use of non-dimensional descriptive terms. Internationalism may be welcomed in technical works as much as elsewhere ; and one of the best ways of encouraging it is to lower the barriers formed by conflicting systems of units. Such hindrances exist even in material intended for English-speaking readers : makers and users of rotodynamic pumps usually think in terms of gallons, but the gallon accepted on one side of the Atlantic is not the same as the gallon on the other side. While not discarding the term *specific speed* which now has so clear a meaning for engineers, the author has proposed an analogous term *shape number.* To some extent the two terms are interchangeable ; but whereas the numerical value of the one may be influenced by the system of units employed and the nature of the liquid flowing through the pump, the value of the other is *invariable* for a given geometrical shape of rotor. A further matter in which clarity has been sought concerns the distinction made between the use of the terms " head " and " energy ". Whenever possible the word *head* is used only in the sense of pressure-head ; an actual quantity that can be measured with a pressure-gauge. Although *energy* in the sense of energy per unit weight can likewise be expressed in terms of feet or metres, yet in this context " feet " is only an abbreviation for " foot-pounds per pound ".

As for the general style of presentation, the author has seen no need to depart from what has been found acceptable in his earlier books. The use of foot units and metric units is a

further indication of the desire to present as impartially as may be the points of view prevailing in Great Britain, America, and Continental Europe. The worked-out Examples, collected in a separate section at the end of the book, will prove the desire to give the most practical possible kind of guidance. It is hoped that the Index will be found complete and informative, thereby encouraging readers to refer to it at once when trying to find their way about the book. As usual, the diagrams —all specially prepared for these books—are intended to illustrate general types of apparatus rather than particular machines; and on this occasion the author would like to acknowledge the help given by Mr. Ramadan Sadek. The Bibliography calls for a word of comment. Many of the papers and articles mentioned therein seem to have very similar titles—titles which in fact do not give a very precise notion of what the paper is about. Although the references (*) in the text of the book itself indicate salient aspects of the papers cited, it is nevertheless to be remembered that an individual paper may have the nature of a valuable and comprehensive small-scale treatise. It is on this account especially that the author is glad to express his indebtedness to the writers concerned.

Some users of this book may think that undue freedom of style has been exercised. In his persistent efforts to make his meaning clear, the author may have strayed too far beyond the boundaries of technical English. It is a risk that has been deliberately accepted. The author would be gratified if his expositions have thereby gained in effectiveness, and he would ask for the forbearance of such readers as are sensitive to overtones or over-enthusiasm.

HERBERT ADDISON.

GIZA, 1948.

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INTRODUCTORY

1. Pumps.—Positive and Rotodynamic.—The definition of a pump as a machine for lifting water is far too restricted for engineering purposes. Machines that are unquestionably pumps frequently do not lift liquids at all, or at any rate they do so only through an insignificant height. Such are boiler-feed pumps, forced-lubrication pumps, booster pumps, and pumps for hydraulic transmission systems. Moreover, the liquid to be pumped may not be water, but may be oil, spirit, milk, sludge, or indeed almost anything that can be made to move along a pipe. A more appropriate definition would therefore be : a pump is a machine which transfers mechanical energy from some external source to the liquid flowing through the machine. That is the essential point : the *energy* of the liquid must be increased. What we do with the energy afterwards in no way concerns the pump. In a waterworks system, for example, the energy is utilised to overcome pipe friction and a gravitational head ; in a fire system the energy is afterwards converted into kinetic energy in the nozzles at the end of the fire-hoses.

Pumping apparatus usually falls into the class of either (i) Positive pumps, or of (ii) Rotodynamic pumps.

Positive Pumps invariably embody one or more chambers which are alternately filled with the liquid to be pumped and then emptied again ; their rate of discharge consequently depends almost wholly on the speed of rotation and hardly at all upon the working pressure. Such machines, e.g. reciprocating ram pumps, gear-wheel pumps, and the like, do not fall within the scope of this book at all. The book is concerned only with—

Rotodynamic Pumps. The essential element of these machines is a wheel or *rotor* of some kind whose rotating blades or vanes impart tangential acceleration to the liquid flowing

through the pump. The motion is continuous, without any of the reversals of direction which characterise a ram pump.

2. Fundamental Distinctions. If we wish to raise a solid object of weight W through a height H, the straightforward way of doing so is to apply a vertical force P equal and opposite to the downward force of gravity W acting on the body, Fig. 1 (i). But it is not the only way. Alternatively, we could apply a *horizontal* force P_1 to the object ; we could get the object moving so briskly that it would mount the ramp and come to rest at the top, Fig. 1 (ii). Although this second proposal looks to be rather an indirect one, it has at least the advantage of flexibility. Whereas in system (i) the lifting

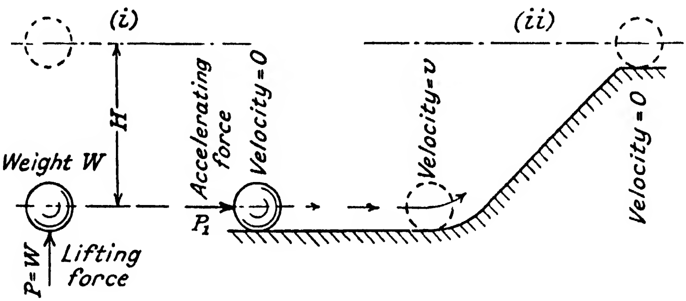


FIG. 1.—Distinction between principles of positive pump (i), and rotodynamic pump (ii).

effort depends strictly upon the weight of the body, in system (ii) there is no essential connection between them. If the effort P_1 is small, it will certainly take a long time to work up sufficient speed to enable the object to climb the slope, and the object will need a long run. But it will get to the top all the same. If, on the other hand, the force available is greater than the weight W, then this vigorous push will impart rapid acceleration and the object will be ready to start climbing very soon after it has begun to move.

Here, then, in the roughest and most general form, is a representation of the basic difference between positive and rotodynamic pumps. In the one, the operating force is applied directly to the liquid, Fig. 1 (i) ; in the other, the force is applied with the intention of generating acceleration in a direction at *right angles* to the general direction of flow, (ii). If we accelerate

a solid or a liquid, we know that we are imparting energy to it ; and that is the whole purpose of the pump.

3. Accelerating the Liquid. On substituting a quantity of liquid for the solid object whose behaviour we have just studied, there will be various methods of showing how acceleration can create a pressure-difference ; though whether these methods will permit us to impart energy to a continuous stream of liquid is another matter.

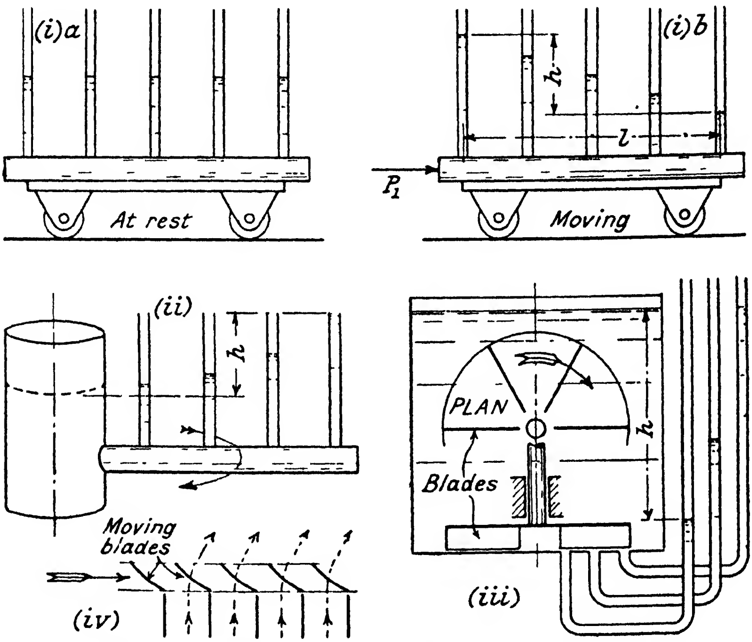


FIG. 2.—Systems of creating a pressure-difference by accelerating a liquid.

(i) *Linear Acceleration.* Here we replace the solid object by a little carriage on which a horizontal closed pipe of length l is mounted, Fig. 2 (i). A set of vertical glass tubes provides the means of observing pressure changes. When the carriage is at rest, the pressure is uniform throughout the length of the pipe, (i) *a* ; but as soon as an accelerating force P_1 is applied to the car, (i) *b*, the liquid tends to lag behind and the liquid in the rearmost gauge-tube stands at a higher level than in the foremost tube. A total difference of pressure-head h has been generated. But it cannot be maintained for very long, for as

soon as the car has reached its safe limiting speed, acceleration must cease and the liquid columns in the gauge-tubes will return to their original uniform level.

(ii) *Radial Acceleration.* By setting the horizontal pipe so that it can revolve about one end, Fig. 2 (ii), radial acceleration will come into play ; in the popular phrase, the liquid is being subjected to centrifugal force. This time the pressure-difference can be maintained indefinitely, and indeed if the uniform speed of rotation is high enough, the apparatus will actually behave like a pump. Liquid will flow out of the outermost gauge-tube, and it will have been lifted through a height h.

(iii) *Forced Vortex Flow.* A better way of subjecting liquid to the effect of centrifugal force is suggested in Fig. 2 (iii). A wheel with radial blades is rotated close to the bottom of the tank ; fixed gauge-tubes show the observer how the pressure-head increases as points further and further away from the axis are studied. Changes in the speed of rotation of the shaft are instantly reflected in changes in the centrifugal head h impressed on the liquid.

(iv) *Use of Curved Vanes.* A general method of imparting acceleration to a stream of liquid is to cause a row of curved blades or vanes to move transversely through it. In Fig. 2 (iv) the liquid is seen moving along parallel passages, until it comes in contact with the blades that cut across its path. Here is a direct example of the *transverse* acceleration mentioned in § 2. The moving blades have pushed the liquid over sideways ; on leaving the blades, it now has a transverse velocity component as well as its original forwards component, and this transverse component can only have been built up as a result of transverse acceleration. Energy has been transferred from the blades to the liquid.

4. Types of Rotor. With a knowledge of the various forms of acceleration to which a flowing liquid may be subjected, it becomes possible to study the shapes of rotor which will most effectively impart the acceleration. Fig. 3 shows in diagrammatic outline the three main types in common use.

They are differentiated by the general or natural direction of flow of the liquid, rather than by the direction of the acceleration impressed on the liquid. As already explained, the two directions are invariably perpendicular one to the other.

(i) *Centrifugal Pump.* Here the wheel generally resembles the bladed element that generated forced vortex flow in the apparatus shown in Fig. 2 (iii). The direction of flow is radially *outwards.* In such machines the rotor is usually termed the *impeller,* Fig. 3 (i).

(ii) *Propeller or Axial-flow Pump.* The incoming liquid approaches the wheel in parallel streams which are themselves parallel with the axis of rotation. As a result of the tangential acceleration that the blades impose on the liquid, the liquid leaving the rotor has a helical or "corkscrew" motion. The

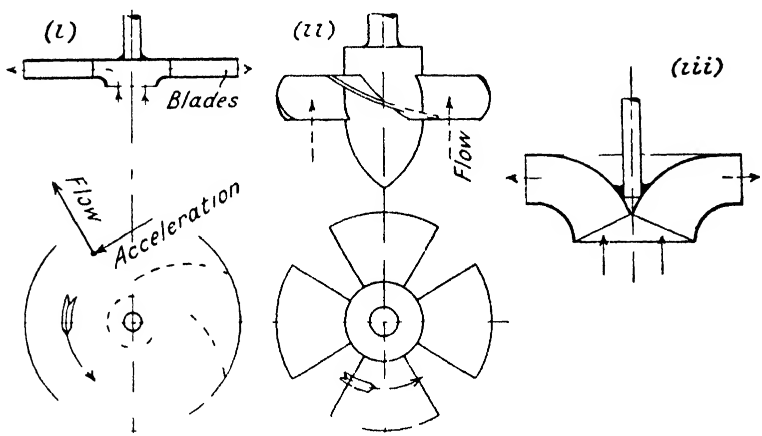


FIG 3 —Types of rotor for rotodynamic pumps

blades have the general shape we associate with a marine propeller, Fig. 3 (ii).

(iii) *Mixed-flow Pump.* Intermediate in shape between types (i) and (ii), the mixed-flow rotor is designed to receive liquid that approaches axially, and it discharges the liquid more or less radially, (iii). Thus the change in direction of flow from axial to radial takes place within the wheel passages.

Although three distinct and clearly-defined shapes of rotor are depicted in Fig. 3, yet in practice there is a continuous gradation of types ranging from pure radial flow to pure axial flow. This makes it difficult and sometimes impossible to place a given rotor in a specific category.

5. General Classification of Rotodynamic Pumps. Centrifugal, mixed-flow, and propeller pumps have so wide a

range of application, and they are built in such a variety of types, that there are many ways of classifying them besides the primary one that depends upon the direction of flow through the rotor. The more usual categories are :—

(i) *Disposition of axis* :—
Horizontal-shaft.
Vertical-shaft.
Inclined-shaft.

(ii) *Number of rotors* :—
Single-stage.
Two-stage, multi-stage, etc.

(iii) *Type of recuperating device* :—
Volute-pump.
Turbine pump, guide-blade pump, diffuser pump.

(iv) *Intensity of pressure generated* :—
Low lift.
Medium lift.
High lift.

(v) *Disposition of casing, inlet and outlet branches, etc.* :—
Side inlet, side suction.
Central suction, balanced suction.
Split casing.
Flange mounted, etc.

(vi) *Materials of construction* :—
All iron and steel.
All gunmetal (or bronze, etc.).
Bronze-fitted (iron casing, bronze shaft and impeller, etc.).
Lead-lined (rubber-lined, etc., etc.).

(vii) *Method of drive* :—
Belt driven.
Direct-coupled.
Gear driven.
Electric-motor driven.
Diesel-engine driven.
Steam-turbine driven, etc., etc.

(viii) *Duty, purpose, or distinctive features* :
General-purpose pump.
Waterworks pump.
Sewage pump.
Irrigation pump.
Drainage pump.
Bore-hole pump.
Boiler-feed pump.
Circulating-water pump.
Condensate-extraction pump.
Bilge-pump.
Ballast-pump.
Fire-pump.
Trailer-pump.
Oil, acid, milk, spirit, paper-stock pump, etc., etc.
Dredge pump.

Self-priming pump.
Hydraulic-storage pump.
Etc., etc., etc.

(ix) *Proprietary titles* :—
Manufacturers often distinguish their own products by giving them trade names—sometimes rather fanciful ones.

Evidently, then, quite a long phrase may be needed to give a comprehensive description of a particular pump; and even then it may not be quite precise. Thus the description *horizontal motor-driven direct-coupled low-lift gunmetal-fitted split-casing pump* may still leave one in doubt as to the form of the rotor. Indeed, the rotor may be so nearly poised on the border-line between the centrifugal type and the mixed-flow type that it might be misleading to classify it as the one or as the other.

6. Scope and Treatment. The range of subject-matter that a reader may expect in a book such as this one can be still better realised when the range of performance of rotodynamic pumps is defined.

Discharges up to 10 tons/sec. per pump are not unusual.
Pressures up to 3000 lb./sq. in. may be generated.
Powers up to 30,000 h.p./unit may be absorbed.

But whatever the size of the pump, the engineer will presumably be interested in one or more of its several aspects, viz.: how it works, how it is built, how it behaves, and how it should be installed; and the various parts of the book have been allocated accordingly.

As users of pumps are much more numerous than makers of pumps, the needs of pump users have especially been kept in mind. Nevertheless, it is hoped that the book will be found useful in design offices. Expositions of principles have intentionally been kept free of involved mathematical analysis; and rules, formulæ, and graphs have been presented in such a way that they can easily be modified to suit the individual experience of the designer. The reader is assumed to have an elementary knowledge of the laws of hydraulics.

PART A

PRINCIPLES

CHAPTER I

THE CENTRIFUGAL PUMP IMPELLER: IDEAL CONDITIONS

7. Basic Conditions. Before we can visualise a pump actually at work, there must be a closed casing or housing for the impeller to revolve in, and inlet and outlet pipes or passages for the liquid. They are shown schematically in Fig. 4. As it is extremely helpful to think first of the natural flow of the liquid through the impeller, § 4 (i), before there is any question of imparting energy to it, the diagram represents the flow under gravity from one reservoir or tank to another one at a lower surface level. The impeller, in fact, is not impelling anything; we may imagine it provisionally as having no blades, but merely circular discs or *shrouds* for guiding the liquid radially outwards. Let it be noted that the liquid is *flowing* into and through the pump. The pump is not lifting or sucking the liquid. Even when for convenience we set the pump above the level of the inlet reservoir, we know that the pump does not really draw the liquid up to it; it passively accepts what the atmospheric pressure has offered it. Moreover, when at a later stage we give the wheel the means of transferring energy to the liquid, this need in no way affect the tranquil flow into the apparatus.

Under the natural conditions suggested in Fig. 4, the difference in surface level between the two reservoirs will be a measure of the energy loss the liquid sustains. Here we may disregard frictional losses in pipes and impeller, and take into account only the eddy loss—destruction of velocity head—as

the liquid issues from the outlet pipe. The conventional system of plotting the hydraulic gradient and the energy line permits such losses to be readily expressed graphically, and this system will be used consistently throughout the book.

8. Control of Flow Velocity. Let it be stipulated that whatever happens to the liquid afterwards, the rate of flow or the discharge into the pump is to be maintained at a constant value q throughout the whole course of the investigation. On

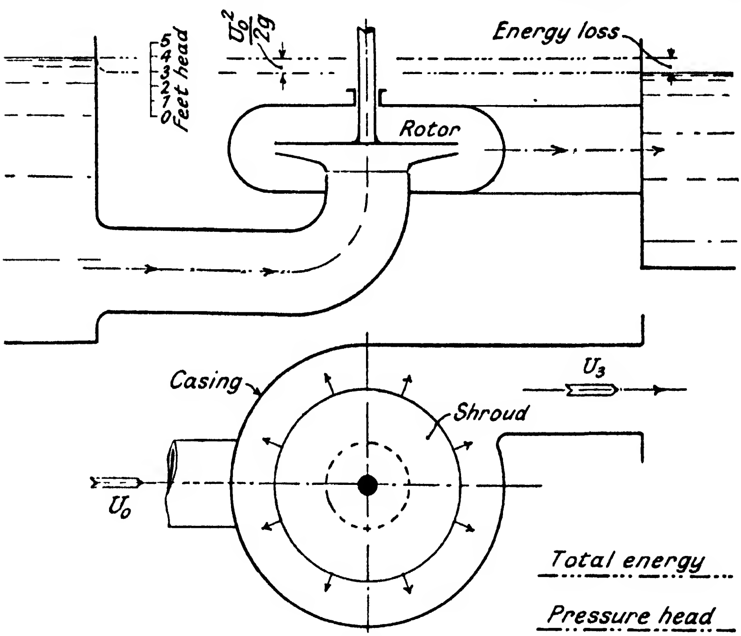


FIG. 4.—Free flow of liquid through passages of elementary pump.

the other hand, the radial *velocity of flow* through the space between the impeller discs may be modified as desired. In general, if

q = discharge through impeller,
r = radius of a selected point,
b = axial distance between shrouds at that point,
Y = radial velocity of flow at that point,

then, since discharge = area × velocity, we have

$$q = 2\pi r b Y, \text{ or } Y = \frac{q}{2\pi r b}.$$

The effect of various alterations in the shape of the impeller shrouds is suggested in Fig. 5. If the shrouds are true discs, set close together as in diagram (i), evidently as an element of liquid moves outwards its velocity is not uniform but continuously falls away, e.g. velocity Y_2 is less than Y_1. As its velocity energy correspondingly decreases, and as energy losses are here ignored, the hydraulic gradient or pressure-head line is seen to *rise*. By spacing the shrouds further apart, as in (ii), velocities are consistently lower. But if, on the other hand, the wheel is of varying width as in Fig. 4, growing narrower towards the outer rim in such a way that the product *br* is constant, then the velocity of flow will be *uniform*, Fig. 5 (iii).

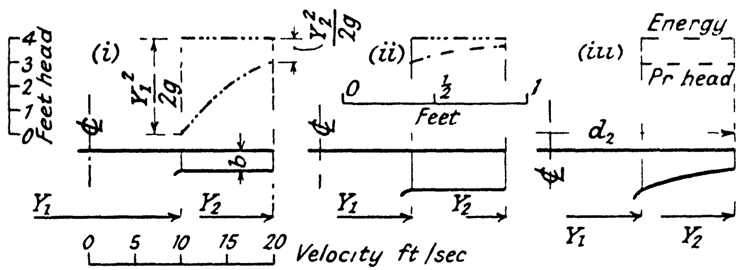


FIG. 5.—Influence of impeller width, etc., on radial velocity. (Uniform impeller diameter = 2 ft.; uniform discharge = 5 cub. ft./sec.)

As in this event the velocity head is also uniform, the hydraulic gradient is seen to be horizontal.

In conditions (i) and (ii) it is to be noted that a negative acceleration is imposed on the liquid as it flows through the impeller. But this acceleration is accompanied by no change of energy, and so has no effect on the subsequent working of the pump. It is *not* tangential acceleration.

9. The Datum Blade. To impart the tangential acceleration to the liquid which alone will transform the apparatus into a pump, there must be a series of blades between the shrouds of the impeller: we must give the wheel something to impel the liquid with. Not any kind of blade will do. There is indeed one particular shape of blade which (in the ideal circumstances now in question) will have no effect on the flow whatever. Although the shaft and the wheel are rotating at the specified speed; although the liquid continues to flow at a rate of discharge q just as it did in Fig. 4, yet the liquid levels remain

just as they were in Fig. 4. The apparatus is still not pumping. The blades seem to behave as shadow or ghost blades—they slip through the liquid quite ineffectively.

Using the shape of wheel that gave constant flow velocity to the liquid, Figs. 4 and 5 (iii), the geometrical form of the "ghost" blades can readily be plotted. The construction is given in Fig. 6. After successive small equal intervals of time, an element of liquid will occupy the successive positions a_1, a_2, a_3. . . . The corresponding positions of a point on the impeller inner rim are shown at b_1, b_2, b_3. . . . Thus in relation to the impeller the liquid element will appear to slide along the heavy curve $b_5 - a'_1 - a_5$, which represents the required blade form. The curve is seen to be an Archimedean spiral; or it is comparable with the profile of a cam designed to give uniform rectilinear motion along a radial line.

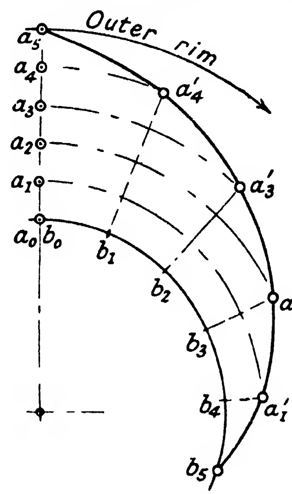


FIG. 6.—Construction for shape of datum blade.

A convenient name for this particular blade form—the form that gives zero tangential acceleration to the liquid—is *datum blade.*

10. The Working Blade. An actual impeller blade, that really does impart tangential acceleration, will have some such shape as is given by the line b_5d_5 in Fig. 7. So far from slipping through the liquid like a ghost, it pushes the liquid vigorously sideways like a man elbowing his way through a crowd. When the liquid enters the impeller at point d_0, it has zero tangential velocity; when it leaves the impeller at point d_5, it has a very sensible tangential velocity. Therefore it must have been subjected to tangential acceleration. Nevertheless the original uniform radial flow component Y has in no way been altered.

Definition of Velocities. Here it is necessary to give distinguishing names to the various velocity components involved:

The original radial component $Y = \dfrac{q}{2\pi br}$ (§ 8) will be termed the *velocity of flow.*

The tangential component imparted to the liquid is termed the *velocity of whirl*, denoted by V.

The *relative velocity* or velocity with which the liquid seems to slide along the blade is denoted by v_r.

The *absolute velocity* or velocity in relation to the fixed casing is denoted by U.

The *peripheral* or *tangential velocity* of a point on the *blade* or impeller itself is denoted by v.

The *angular velocity* of the impeller or rotor is ω, and the *rotational speed* in revolutions per minute is N.

A comparison between the datum blade (reproduced from Fig. 6) and the working blade, Fig. 7, provides the means of establishing the *absolute path* of the liquid flowing through the wheel. At a given radius the circumferential distance between the two blades must evidently represent the circumferential distance through which the working blade has pushed the liquid sideways; thus $a'_3c_3 = a_3d_3$, etc. If the points plotted in the diagram relate to successive small intervals of time t, then at a selected radius we can write down by direct measurement, say :—

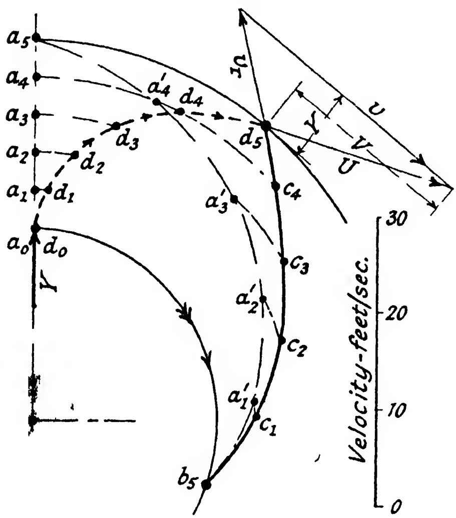


FIG. 7.—Diagram showing :-
Datum blade $b_5\ a'_1\ a'_2\ a'_3\ a'_4\ a_5$
Working blade $b_5\ c_1\ c_2\ c_3\ c_4\ d_5$
Absolute path of liquid $d_0\ d_1\ d_2\ d_3\ d_4\ d_5$

$$Y = \overline{a_2a_3}/t,$$
$$U = \overline{d_2d_3}/t,$$
$$v_r = \overline{c_2c_3}/t.$$

Thus, a comparison between the distance $d_0\ d_1$ and the distance $d_4\ d_5$ convincingly shows how the absolute velocity of the liquid has steadily *increased* during its passage through the wheel. The relation between the velocity of flow, the velocity

of whirl, the absolute velocity, and the peripheral rotor velocity can be found by a direct graphical or vectorial method. If, for instance, conditions at the rotor outlet are to be considered, we first choose a suitable scale of velocities, and plot v_r and U tangentially respectively to the blade tip and to the absolute path. The construction is made clear in Fig. 7. The "vector triangle" or "velocity triangle" is a convenient way of stating the fact that the relative velocity is the vector difference between the rotor velocity and the absolute velocity of the liquid.

11. Energy Equations. The effects on the pump performance of substituting "working" blades for "datum" blades are very marked (*).† They are: (i) the shaft and rotor no longer seem to spin round idly, of their own accord, but on the contrary they offer a very positive resistance. Quite a definite torque must be exerted on the shaft: energy must continuously be fed into it; (ii) the liquid will undergo an increase of pressure as it flows through the pump. In short, the pump at last is behaving like a pump. If the acceleration given to the liquid were linear, as it was in Fig. 1 (ii) or Fig. 2 (i), it would be easy to evaluate the energy input. If the moving object had unit weight, then to accelerate it from rest to a velocity u would need an energy input of $\frac{1}{2}u^2/g$, viz.: $\frac{1}{2}\dfrac{u \cdot u}{g}$, where g is the acceleration of gravity. But the evaluation of tangential acceleration is more complex, and may more conveniently be studied in Chapter II, § 18. The desired energy equation applicable to circular motion can preferably be derived from a study of changes of angular momentum. At inlet to the rotor the liquid has no tangential velocity component and therefore no angular momentum, Fig. 7. At exit from the rotor, at a point at radius r where the whirl component is V, the angular momentum per unit weight per second is Vr/g.

If W is the weight of liquid per second flowing through the wheel, then change of angular momentum per second = $\dfrac{W}{g} \cdot Vr$, which is equal to the torque exerted on the wheel. Also energy input to wheel per second = torque × angular velocity

† The asterisk (*) denotes an item in the Bibliography, page 481.

$$= \frac{W}{g} . Vr . \omega = \frac{W}{g} . Vr . \frac{v}{r}$$

$$= \frac{W}{g} . Vv . \qquad . \quad . \quad . \quad (1\text{-}1)$$

or energy input per unit weight of liquid

$$= \frac{Vv}{g} \qquad . \quad . \quad . \quad . \quad (1\text{-}2)$$

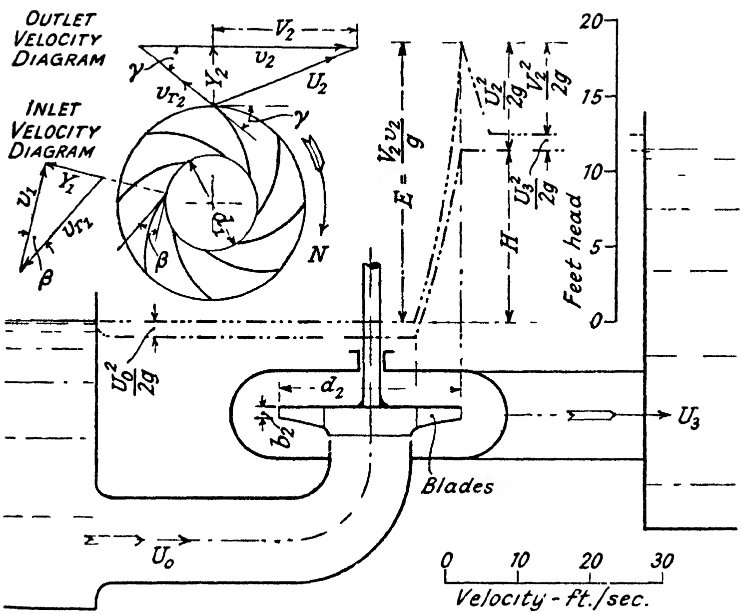


Fig. 8.—Changes of pressure-head and energy created in ideal centrifugal pump by impeller as in Fig. 7. 2 ft. diam., delivering 5 cub. ft./sec.

This is one form of the Eulerian equation of energy ; its similarity to the equation for linear acceleration, above, is very evident.

12. Graphical Plotting of Energy. The manner in which the liquid utilises the energy the rotor has transferred to it is depicted schematically in Fig. 8. This diagram is identical with Fig. 4, except for the changes created by the blades : the rate of discharge q or the weight of liquid per second W remains *unaltered*. Considering a datum plane which includes

the liquid surface in the inlet tank, we can assess as follows the energy per unit weight of the liquid leaving the impeller :—

Pressure energy per unit weight of liquid $= H$
(where H is the *head* or difference in level between the two tanks)

Velocity energy . . . . . $= \dfrac{U_2^2}{2g}$

Equating energy input E to energy output, we can write :—

$$E = \frac{V_2 v_2}{g} = H + \frac{U_2^2}{2g} \quad . \quad . \quad \textbf{(1-3)}$$

(*Note*.—In order to distinguish conditions at different points in the pump, the following subscripts are used :—

(0)	refers to	conditions	at entry to the pump casing
(1)	,,	,,	at entry to the rotor.
(2)	,,	,,	at outlet from the rotor.
(3)	,,	,,	at outlet from the casing)

Here it is useful to study the various changes that the liquid undergoes during its journey from the inlet tank to the outlet tank. Starting with zero energy, the liquid has to acquire velocity energy $\frac{U_0^2}{2g}$ in the inlet pipe by borrowing from its pressure or its position energy. At the impeller entrance the total energy is still zero, and the velocity energy is $\frac{Y_1^2}{2g}$, where Y_1 is the velocity of flow. As the liquid is now swept out of its course by the sideways thrust of the impeller blades, its absolute velocity rises (Fig. 7), its pressure rises, and thus its total energy rises still more rapidly. Of the velocity energy $\frac{U_2^2}{2g}$ that the liquid has acquired by the time it leaves the impeller, a quantity $\frac{V_2^2}{2g}$ is wasted by eddying in the casing, and the remainder $\frac{U_3^2}{2g}$ is thrown away by eddying in the outlet tank. (It is here assumed that $U_0 = Y_1 = Y_2 = U_3$.)

The drop in the energy line in Fig. 8 already suggests a serious fault in the pump performance. It is impossible to apply tangential acceleration to the liquid without increasing its velocity; yet apparently the energy equivalent to the

increase in velocity is completely lost. In fact it is not essential to dissipate this energy ; it may partly be recuperated by the methods described in Chapter IV. But for the moment we shall continue to study the elementary form of pump shown in Figs. 4 and 8.

Other information to be found in Fig. 8 includes the graphical construction for determining the inlet blade angle β and the outlet blade angle γ ; the one is taken from the "inlet velocity triangle", and the other from the "outlet velocity triangle". The latter is in this instance similar to the velocity triangle plotted in Fig. 7. The former embodies the three vectors :—

v_1 = inner rim velocity,
Y_1 = velocity of flow at impeller entry,
v_{r1} = relative velocity of liquid at impeller entry.

13. Derived Equations. For practical purposes we require modified forms of the fundamental equation 1-3 above. It is found preferable to eliminate the terms U and V, and to introduce a term descriptive of the *blade shape*. The usual way of doing so is to utilise the *outlet blade angle* γ, viz. : the angle between a tangent to the rotor rim and a tangent to the blade, at their point of intersection (Fig. 8). From the velocity diagram there reproduced it is clear that

$$U_2^2 = V_2^2 + Y_2^2$$

and that

$$V_2 = v_2 - Y_2 \cot \gamma.$$

Inserting these values in equation 1-3 results in the equation

$$H = \frac{v_2^2 - Y_2^2 \operatorname{cosec}^2 \gamma}{2g} \quad . \quad . \quad (1\text{-}4)$$

which gives an ideal value of the height through which the liquid could be lifted by the impeller alone.

The ideal *efficiency* can likewise be expressed in similar terms. Using for it the symbol η_i, we can write :—

$$\eta_i = \frac{\text{useful energy output per second}}{\text{total energy input per second}} = \frac{W \,.\, H}{W \,.\, \dfrac{V_2 v_2}{g}} = \frac{gH}{V_2 v_2} \quad (1\text{-}5)$$

Substituting the values of H and V_2 derived above, we find that

$$\eta_i = \frac{v_2^2 - Y_2^2 \operatorname{cosec}^2 \gamma}{2v_2(v_2 - Y_2 \cot \gamma)} \quad . \quad . \quad . \quad (1\text{-}6)$$

The ideal *discharge* of the impeller, q, can be put in terms of the flow velocity and the impeller dimensions. If d_2 is the outside diameter, and b_2 the axial width, then $q = \pi d_2 b_2 Y_2$.

The relation between *rim speed* and *speed of rotation* is :—

$$v_2 = \frac{\pi d_2 N}{60}$$

The ideal *power output* or water horse-power (W.H.P.) is $\frac{WH}{k_p}$, where k_p represents energy per second corresponding to one horse-power.

The ideal *power input* or shaft horse-power (S.H.P.), is expressed by $\frac{\text{W.H.P.}}{\eta_i}$. **(Example 1) †**

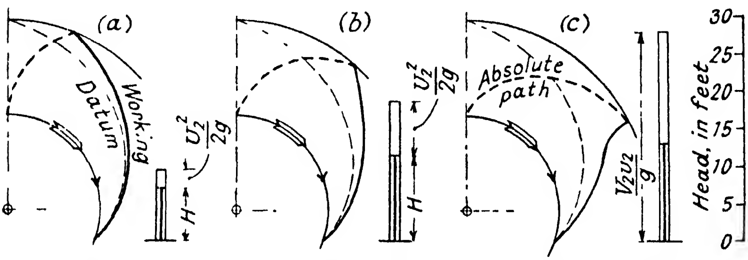


FIG. 9.—Influence of blade shape on impeller performance for constant peripheral speed.

14. Relation between Blade Form and Impeller Performance. There are two instructive ways of observing how changes in the blade shape may influence the behaviour of the rotor :—

(i) With *constant* rotational speed, *constant* impeller diameter, and *constant* flow velocity and rate of discharge, we can note how the ideal *head* varies as the shape of the working blade departs more and more widely from the shape of the datum blade (§ 9).

(ii) With *constant* head and *constant* impeller diameter, we can trace a connection between *blade angle, speed*, and ideal *efficiency*.

Method (i) : For these conditions the original datum blade drawn in Fig. 6 will serve. It is reproduced in Fig. 9, together

† See Part E, page 427.

with three shapes of working blades (*a*), (*b*) and (*c*), designed to give outlet whirl velocity components respectively of 10, 20, and 30 ft./sec. The formulæ of § 13 permit the corresponding ideal heads to be computed, and to be recorded in the diagram. Just as we might have expected, the blade form that most forcefully pushes the liquid out of its original radial path is the one that generates the highest head H. The greater the curvature of the absolute path of the liquid, the greater the energy required (*).

Method (ii): To show the value of a non-dimensional approach to the problem (§ 52), we may here adopt a general treatment in which velocities are expressed as a ratio of the *spouting velocity*, $\sqrt{2gH}$. It is the ideal velocity at which

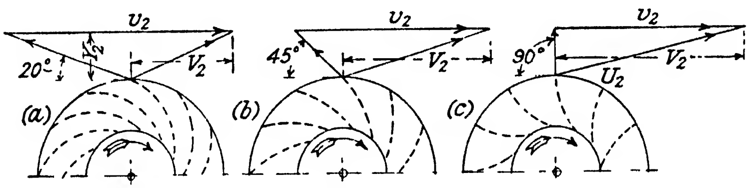


FIG. 10.—Influence of blade shape on velocity diagrams: head and impeller diameter constant.

liquid would issue from an orifice under an ideal head *H*. Choosing a velocity of flow of $\frac{1}{4}\sqrt{2gH}$, and outlet impeller blade angles respectively of $\gamma = 90$ degrees, 45 degrees, and 20 degrees, then from the expressions in § 13 we may compute the values of speed and ideal efficiency as follows: –

Ref	Outlet Blade Angle γ	Peripheral Speed $\dfrac{v_2}{\sqrt{2gH}}$	Ideal Efficiency
(*a*)	20 deg.	1·24	0·73
(*b*)	45 ,,	1·06	0·58
(*c*)	90 ,,	1·03	0·47

The equivalent impellers and outlet velocity triangles are drawn to scale in Fig. 10. Here is information to reinforce what was supplied in § 12 and in Fig. 8. The low apparent efficiency associated with a large outlet blade angle is evidently related to the high value of the absolute outlet velocity U_2 that

is clearly shown in the diagrams, Figs. 9 (*c*) and 10 (*c*). The wheel is throwing away far too much energy, instead of turning it into useful pressure head. The need for the recuperation devices to be described in Chapter IV becomes still more manifest.

15. Alternative Expressions for Pressure-rise in Impeller. Imagine the impeller to be at rest. Flow is maintained through the passages at the normal rate, and thus the conditions in one such passage (bounded by a pair of blades and by the two shrouds) are as indicated in Fig. 11. A direct application of Bernoulli's theorem provides the value of the corresponding pressure-head difference, thus :—

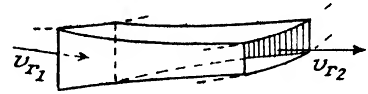


FIG. 11.—Shape of one impeller passage.

Gain in pressure-head
= loss of velocity energy

or
$$h_{gi} = \frac{v_{r1}^2}{2g} - \frac{v_{r2}^2}{2g}.$$

Now let the wheel be run up to speed, the flow velocity being maintained at its invariable value Y. The forced-vortex motion now impressed on the liquid (§ 3 (iii)) will generate centrifugal head on the liquid, of an intensity :—

$$h_c = \frac{v_2^2}{2g} - \frac{v_1^2}{2g}.$$

The total difference of pressure-head generated by the impeller under working conditions, as the liquid traverses a passage, will therefore be :—

$$h_i = h_{gi} + h_c = \frac{v_{r1}^2 - v_{r2}^2 + v_2^2 - v_1^2}{2g}.$$

By means of the substitutions in § 13, it follows that

$$h_i = \frac{v_2^2 - Y_2^2 \operatorname{cosec}^2 \gamma}{2g} + \frac{Y_1^2}{2g} = H + \frac{Y_1^2}{2g}.$$

The reason why the pressure-head difference h_i is greater than the ideal head H is manifest from Fig. 8 ; just before entering the impeller the pressure-head of the liquid is *lower* than the

datum level by an amount $Y_1^2/2g$, which is here assumed to be equal to $\frac{U_0^2}{2g}$.

Although this treatment is illuminating, it lacks the sense of vigour of the earlier method (§§ 11-13). There is nothing to suggest that the blade passages are not drifting quietly around, whereas in fact they are being very resolutely *driven* round.

CHAPTER II

ACTUAL FLOW CONDITIONS IN THE IMPELLER

16. What really happens in the Impeller ? Thus far we have studied no more than an idealistic outline of a centrifugal pumping apparatus. The disillusioning process of tracking down flaws in the theory must now begin. There are several obvious ones. (i) In the inlet and outlet pipes, Fig. 8, friction and the like will certainly steal energy from the flowing liquid. (ii) Similar losses will occur in the wheel passages themselves—the passages such as are shown in Fig. 11. (iii) The flow through the impeller will not be strictly two-dimensional if the dished type of impeller hitherto studied is used (Figs. 4 and 8). (iv) The blades must be of metal of finite thickness ; their thickness cannot be regarded as negligibly small. Modifications to velocities and velocity diagrams may thus be necessary. (v) Before the liquid enters the impeller passages at all, there is a chance that the liquid streams may have been influenced by the proximity of the moving blades—it may have acquired an initial velocity of whirl.

But the fundamental uncertainty lies here : Have the rotor blades in fact impressed on the liquid the tangential acceleration we intended them to do ? Only by getting into the closest sympathy with the liquid can we guess what the answer is likely to be. The liquid does not want to be jostled and bustled. It wants to keep on flowing tranquilly along its outward radial path. So naturally when the impeller blades threaten it with rough and even violent treatment, the liquid will take whatever evasive action that is open to it. Suppose we try to impart tangential velocity components to the liquid in a tea-cup.

When we use a tea-spoon as a means of applying energy, we can watch the liquid trying to escape round the edges—to leak away sideways—to do anything rather than submit to being stirred or pushed round. In a high-pressure centrifugal pump the treatment will be very much more severe. Sometimes the whirl component impressed on the liquid is comparable with the speed of a golf-ball ; and the liquid is compelled to attain this tangential velocity—which may exceed 150 ft./sec.—from a standing start in a period of time that may be less than $\frac{1}{50}$ sec. One must never forget the *violence* that often accompanies the process of energy transfer.

17. The Experimental Evidence. Corrections to the basic theory called for by factors (i) to (iv) above are discussed in the chapters of the book devoted to design and performance. In regard to (v), experiments with injected streams of dye suggest that if the liquid streams do acquire a whirl component before entering the impeller, it is only to an insignificant extent (*). Unless such initial whirl is deliberately encouraged, then, § 19, it may usually be assumed that the liquid enters the wheel radially.

When comparing the actual whirl component imparted by the blades with the ideal component as computed by diagrams such as Fig. 8, it is convenient to use the symbols :—

V_∞ = ideal outlet tangential velocity (hitherto denoted by V_2.)
V_n = actual component.

The general line of thought followed in the previous paragraph has prepared us to believe that these values are not equal. Experimental evidence fully justifies this belief. (*a*) If in an actual pump the head generated is measured, and the ideal value of energy input is computed, viz. $\frac{V_\infty v_2}{g}$, then the disparity between the two is greater than can be accounted for by the most generous allowance for energy losses of all kinds. (*b*) Similarly if the power input to the pump is measured, it would be found insufficient to generate a tangential velocity of V_∞. (*c*) A special technique of measurement, by which the magnitude and direction of the absolute exit velocity U_2 can be determined (*), shows that the relative angle at which the liquid leaves the

wheel is *less* than the nominal blade angle γ. (*d*) Experiments made under identical conditions except for the number of blades in the impeller give these results : if the blades are fewer than perhaps 8 or 6, then the head generated falls away as the blade number is reduced, although the speed of rotation and the rate of flow remain unaltered. If still more blades were taken away— if only one blade remained—then it would tend to churn up the liquid fruitlessly instead of driving it round.

18. Evaluation of Tangential Acceleration. In trying to explain these discrepancies it is profitable to find a way of actually evaluating the tangential acceleration—a problem that has been lying side-tracked since § 11. The problem is not intractable if we first study a single solid object before passing on to streams of liquid elements. In Fig. 12 the object is seen sliding with uniform linear velocity Y along a straight rod which rotates about a fixed centre with uniform angular velocity ω. Although neither of these motions considered separately can create tangential acceleration, yet in combination they produce all the effects that we associate with acceleration. There is a very positive means of demonstrating this. If we simulate the rod in the diagram by swinging a walking-stick round and round, hardly any effort is needed to maintain the motion. But if we now slip a heavy iron washer on to the stick, and allow the washer to slide outwards as the stick revolves, quite a sensible retarding or braking effect will be noticed. A torque must be applied to the stick to maintain its revolutions ; and the tangential velocity with which the washer is finally released gives a very convincing measure of the energy that has been imparted to it. But if friction be disregarded, the only force that can have been applied to the washer is a *tangential* one, viz. a force P perpendicular to the stick.

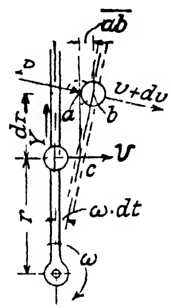


FIG. 12. - Solid object undergoing Coriolis acceleration.

Returning now to the ideal diagram, Fig. 12, we may say : At radius r, the object has a tangential velocity v. After a small interval of time dt, during which the object has moved along the rod a distance dr, the tangential velocity has increased to $v + dv$.

Now if the rod had not forced the object sideways—if the rod had been hinged at point c like a flail, so that the part ca moved parallel to itself—then at the end of the interval dt the object would only have arrived at point a instead of at point b. Since the angular distances traversed by the rod in time dt is ωdt, evidently the linear distance ab is

$$dr \,.\, \omega dt = Y \,.\, dt \,.\, \omega dt.$$

Denoting the tangential acceleration by the symbol α, and inserting values in the Newtonian equation

$$\text{Distance} = \tfrac{1}{2}(\text{acceleration}) \times (\text{time})^2,$$

we have $\overline{ab} = \frac{1}{2}\alpha(dt)^2$

or $Y \,.\, dt \,.\, \omega dt = \frac{1}{2}\alpha(dt)^2$

from which :—

$$\textit{Tangential acceleration } \alpha = 2Y\omega \quad . \quad . \quad \textbf{(2-1)}$$

In Continental Europe this particular form of acceleration is known as Coriolis acceleration, from the French mathematician of that name. Only the simplest example of it has here been studied.

19. Ideal Pressure-distribution in Rotor Passages. To reproduce in an impeller the simple radial motion postulated in Fig. 12, the blades must be radial likewise. Such a one is depicted in Fig. 13 ; otherwise the conditions are assumed to be identical with those of Figs. 4 and 8. Fixed inlet guide blades may be imagined for imparting to the liquid an initial whirl component equal to the inner rim velocity v_1, thus suppressing eddy losses at inlet. If two rows of glass gauge tubes could be set up as indicated, a radial row cc and a circumferential one bb, and if some sort of stroboscopic device could be contrived for observing them as they moved round, then we should see the liquid columns disposed as shown in the upper views. Because of the *radial* acceleration impressed on the liquid, the pressure-head increases from the inner to the outer rim of the wheel, § 3 (iii) ; because of *tangential* acceleration, the pressure-head increases from the back of one blade to the front of the next blade. It is highly instructive to note that tangential acceleration has created just the same type of pressure diagram that we have already seen associated with linear acceleration, Fig. 2 (i) (b).

When computing these pressure variations analytically, we first study a small element of liquid enclosed between a pair of blades and the side plates of the impeller, as distinguished by hatching in Fig. 13. Its characteristics are : Radial thickness $= dr$; circumferential length $= l = 2\pi r/n$, where n is the number of radial blades ; axial width $= b$; volume $= l \,.\, b \,.\, dr$; weight $= w \,.\, l \,.\, b \,.\, dr$; radial velocity $= Y$. Since the normal law, force = mass × acceleration, still applies, evidently the tangential force required to give the element its tangential

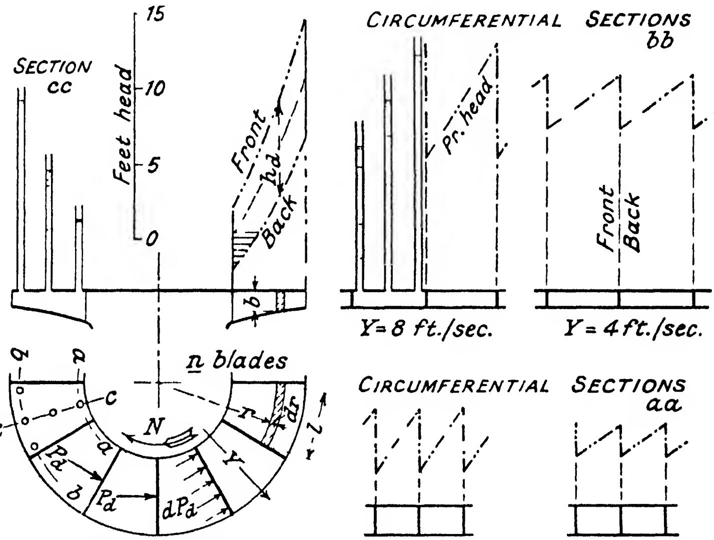


FIG. 13.—Effect of tangential and radial acceleration on pressure-distribution in radial-bladed impeller.

acceleration $2Y\omega$ (equation 2-1) will be : $w/g \,.\, l \,.\, b \,.\, dr \,.\, 2Y\omega$. Now the only way in which this force can be transmitted from the blades to the liquid is by means of the pressure-difference prevailing at the two ends of the element ; this pressure-difference will have the value :—

$$\frac{\text{force}}{\text{area}} = \frac{\dfrac{w}{g} \,.\, \dfrac{2\pi r}{n} \,.\, b \,.\, dr \,.\, 2Y \,.\, \dfrac{2\pi N}{60}}{b \,.\, dr}$$

$$= \frac{w}{g} \,.\, \frac{8\pi^2}{60} \,.\, \frac{rYN}{n}.$$

The corresponding value of *differential pressure-head* h_d will be :—

$$h_d = \frac{8\pi^2}{60} \cdot \frac{rYN}{gn} \quad . \quad . \quad . \quad (2\text{-}2)$$

It is manifestly identical with the difference between the pressure-heads at the front and at the back of a blade at the given radius.

20. Differential Pressure-head. Equation 2-2 clearly shows that for the ideal conditions stipulated, the differential pressure head h_d prevailing at any radius in the given impeller is subject to the following influences :—

(i) It increases uniformly from the inner to the outer rim of the wheel.

(ii) It varies directly as the radial velocity of flow, viz., as the quantity of liquid flowing through the rotor.

(iii) It varies *inversely* as the number of blades, n.

Some of these effects are suggested pictorially in Fig. 13, where pressure-heads at various points are plotted vertically. It is assumed that along the radial centre-line of the wheel passages the pressure-head is controlled by the laws of forced vortex flow, § 3 (iii) ; while at a given radius the head rises or falls above the mean value in accordance with equation (2-2). In Fig. 13 the ordinates relate both to the original rate of flow of 8 ft./sec., and to a reduced flow of 4 ft./sec. In Fig. 14 the complete enveloping surface is sketched, giving the effect of a solid having the shape of a rather odd kind of milling-cutter.

A further analysis of the ideal pressure-distribution diagram will yield the value of the total dynamic thrust P_d that each blade exerts on the liquid. Alternatively, if we multiply each element of the tangential force dP_d by the radius at which it acts, Fig. 13, the result will be expressed as a torque ; and by integrating between the inner and outer radii of the wheel, the total torque that the wheel must apply to the liquid will result. Finally, total torque × angular velocity = work done on liquid per second. But we already have a measure of the energy impressed on the liquid per second ; it has the value

$$W \, . \, E = W \, (V_2 v_2/g - V_1 v_1/g), \text{ § } 13.$$

The fact that there is exact agreement between values computed by the two methods confirms the underlying principles.

(Example 2)

21. Modified Diagrams. The diagrams reproduced in Figs. 13 and 14 are still only ideal ones; they have been constructed according to assumptions that experiment shows to be unjustified, § 17. Yet they are useful in exposing fairly clearly where errors may lie. There is a dubious look about the parts of the diagrams that purport to represent conditions at the inlet and outlet tips of the blades. Just before entering the blade

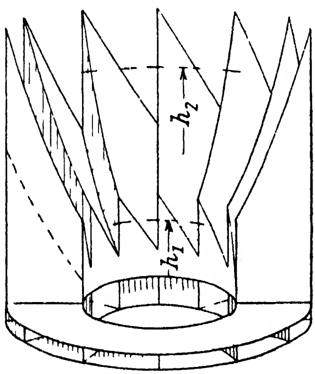


FIG. 14.—Idealised representation of distribution of pressure head, corresponding to Fig. 13, radial velocity $V = 8$ ft. sec.

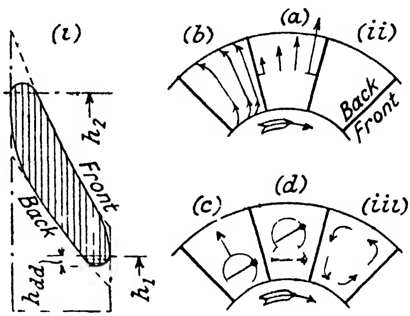


FIG. 15.—Suggestions for plotting true pressure and velocity distribution in rotor passages.

passages, the liquid was assumed to have a mean pressure-head of h_1; yet immediately after entering the passages, the effect of the differential pressure-head is to cause the total pressure-head to rise or fall below this mean value. How can the liquid be induced to accept almost instantaneously these changes?

Another point of criticism is this: if the diagrams are indeed correct, and if the differential pressure-head persists right up to the blade tips, then surely the liquid's natural gift for evasion will enable it to leak round the blade tips, from the high-pressure to the low-pressure zone. We can be pretty sure, then, that the true diagram will be more likely to resemble Fig. 15 (i) than Fig. 14; that is, the pressure-difference fades away to nothing at the blade tips.

22. Actual Pressure and Velocity Distribution. An important consequence follows the disclosure of the reduction in area of the pressure diagram, Fig. 15 (i). The method suggested in § 20 for assessing the work done on the flowing liquid showed very distinctly that there was a direct relation between the area of the pressure diagram and the energy given to the liquid. If the corrected diagram has a smaller area than the ideal one, then the real energy given to the liquid is less than the assumed value E. It follows that the real value of the tangential velocity component V_n is *less* than the ideal value V_∞ : which is just what experiment proves, § 17.

We proceed to question another assumption underlying Figs. 13 and 14, viz., the assumption that the radial velocity of flow Y was uniform at all points at a given radius. Let us trace the distribution of *energy* along the hatched element of liquid shown in the plan view, Fig. 13. The pressure-head or pressure energy increases continuously as we proceed from one blade to the next ; the velocity energy is necessarily uniform because the velocity is uniform ; thus the *total* energy must necessarily vary from point to point. On the other hand, since each element of liquid at a given radius has received the same tangential acceleration, there should be *uniform* energy at all points at that radius. There is a flat contradiction between the two results.

One way of resolving the disagreement is to abandon the belief in uniform radial velocity and to admit that near the back of a blade the flow velocity through the passage is greater than it is near the front of a blade ; the resulting variation in velocity energy might then just compensate for the undoubted variation in pressure energy. The accompanying flow pattern would then have the form sketched in Fig. 15 (ii) (*a*) ; the crowding together of the liquid streams near the back of the blade (ii) (*b*) suggests the relatively greater flow velocity in that region.

23. Theory of Counter-rotation. Another line of approach also supports this hypothesis of non-uniform velocity distribution. It depends upon another manifestation of the liquid's stubbornness : of its refusal to budge if it can possibly help it. The circular element of liquid selected in Fig. 15 (iii) (*c*) cannot help being swept sideways by the rotor blades and

thrust outwards by the bulk flow through the wheel; but it refuses to be twisted round as well. If viscous forces are disregarded, there seems to be no means of rotating the element about its own axis. A diameter that is originally pointing (say) north-east will retain that orientation. During the interval of time required to bring the element from position (*c*) to position (*d*), one end of the diameter will travel a greater distance *relatively to the rotor passage* than the other end, which exactly agrees with the type of velocity variation depicted in Fig. 15 (ii) (*a*).

A gramophone can be used to demonstrate very convincingly this effect of counter-rotation. On the turntable is set a little circular tin-lid containing a few millimetres depth of water: this simulates the circular element of diagrams (iii) (*c*). A match-stalk floated on the water represents the diametral line. When the turntable is cautiously released and begins to revolve slowly, the match is seen to hold its direction—at least during one revolution—almost as insistently as a compass-needle. Although in relation to the room the match has not changed its orientation, yet in relation to the turntable it has rotated in the opposite sense. Applying this analogy to the liquid in the wheel passages, we may accept the view that there is a circulation impressed upon it, complementary to the general outward flow. This secondary motion, suggested in Fig. 15 (iii), not only accounts for the greater radial velocity near the back of the blades, but it also offers a reason for the reduction in the outlet whirl component V_n as compared with the ideal value V_∞. The difference between the two velocities might be accounted for by the backward tangential component of the secondary circulation.

24. Influence of Number of Blades. (*) When advancing from the simple radial-bladed wheel, Figs. 13-15, to the type of impeller with curved blades used in actual centrifugal pumps, there is no reason to abandon the conclusions already established. We may accept for this impeller also the general picture of pressure and velocity distribution presented in Fig. 15. Diagram (i) in Fig. 16 shows how the pressure-head steadily rises as we proceed from the back of one blade to the front of the next; diagram (ii) shows that velocity variations have an opposite tendency. Comparative outlet velocity

triangles show the distortion that results from using the true value of outlet tangential component, V_n, in place of the ideal component, V_∞. The way in which the distortion is linked up with the *number of blades* will now be apparent, too. If this number were infinitely great, none of the influences studied in §§ 22, 23 could have any effect; the liquid would be so positively guided that the velocity diagram (shown in broken lines in Fig. 16) could not but conform to the blade angles of the impeller. As the blades become progressively fewer, so does their control relax until in the end they can do no more than impart to the liquid the tangential velocity V_n which is less than the ideal value V_∞ by an amount x.

The significance of the distinctive subscripts $_\infty$ and $_n$ is now apparent; V_∞ is the tangential velocity that an infinite number

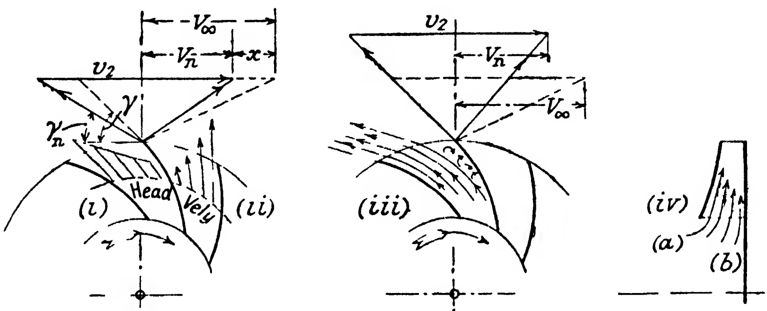


FIG. 16.—Comparison between ideal and actual impeller performance.

of blades would impart, while V_n is what in fact the finite number n imparts. In pump design it is naturally a matter of the highest importance to be able to estimate the ratio between the two, § 93. If we know that a value V_n would correspond with the actual energy increment we desire to give to the liquid flowing through the rotor, then we design the blade outlet angle γ in accordance with the *ideal* value V_∞. The difference between the angles γ and γ_n is the practical expression of the whole of the arguments developed in the preceding paragraphs. An impeller based on the blade angle γ might reasonably be expected to generate the head we expected of it; an impeller based on the angle γ_n might only create a head equivalent to 60 or 70 per cent. of our expectations. In

other words, the true relative velocity of the liquid leaving the impeller is *not* tangential to the tips of the impeller blades.

25. Some Other Possibilities. There remains an alternative theory to explain the discrepancy x between the values V_∞ and V_n. It assumes that the relative stream in the impeller passages breaks away from the back of the blade as suggested in Fig. 16 (iii), leaving zones of dead liquid near the rotor outlet. Since this live stream now has a reduced effective cross-sectional area, its relative velocity will be raised above the hitherto-accepted value, and the resulting effect on the outlet velocity triangle is clearly indicated. Quite apart from any question of counter-rotation or the like, § 23, the corrected whirl component V_n will be less than the ideal value. It is highly probable that this suggested type of flow disturbance actually does occur in conditions of reduced discharge, § 204.

Fig. 16 (iv) serves as a reminder that we have been on very doubtful ground in continually assuming that, while under the influence of the impeller blades, the liquid elements always move each one in a plane perpendicular to the rotor axis. Manifestly in the impeller shown in the diagram such a supposition will not be true. As the liquid flowing into the wheel passages alters its direction from an axial course to a nearly radial course, it seems probable that a kind of free vortex motion may result, with the consequence that the relative velocity will be greater at point (a) than it is at (b). Corresponding additional complexities in the pressure-distribution over the blades may thus be expected, § 33.

26. Average Blade Loading. By this time it should be fairly clear that it would be very difficult to construct, for an actual impeller with curved blades, the pressure diagrams that in Figs. 13 and 14 served for ideal impellers with radial blades. Yet some knowledge of what the differential pressure-head is likely to be is so useful that even a rough estimate is better than nothing at all. Such an estimate might be made thus :—

Let P_d represent the total resultant dynamic thrust on the blade, Fig. 17,

P_t = the tangential component of this thrust,

a_d = the area of the blade projected normally to P_d,

a_t = the area of the blade normal to P_t, viz. the area projected on a diametral plane, as shown by hatching in the diagram,

r_g = the radius of the centre of gravity of the area a_t, Fig. 17,

P_w = the horse-power delivered to the liquid. This may be taken as $\frac{W}{k_p} \cdot \frac{V_n v_2}{g}$,

T = torque exerted on each of the n blades, $= P_t \cdot r_g$.

Now work done per second on the liquid $= P_w \cdot k_p$. But work done per second also = total torque $\times$ angular velocity. Therefore

$$P_w \cdot k_p = nT \times 2\pi N/60,$$
$$= n \cdot P_t \cdot r_g \times 2\pi N/60,$$

from which the value of P_t may be extracted.

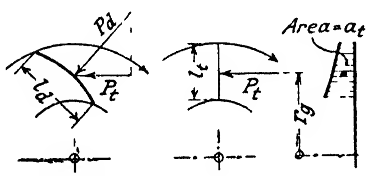


FIG. 17.—Estimation of dynamic thrust on a blade.

Also the distances l_d and l_t in Fig. 17 may be taken to represent respectively the areas a_d and a_t, from which it follows that

$$\frac{P_d}{P_t} = \frac{l_d}{l_t} = \frac{a_d}{a_t}.$$

As P_t is already known, we can say that :—

$$\text{Average differential pressure-head} = h_d = \frac{P_d}{a_d} = \frac{P_t}{a_t}.$$

As the value so obtained does not profess to be more than a rough approximation, its utility for comparative purposes will be improved if it is expressed in the form of a ratio—the ratio between the average differential head and the total head generated by the pump, H.

In this book the ratio will be termed the *relative blade loading* or simply the *blade loading*. Denoted by the symbol ϵ, it has the value :—

$$\epsilon = \frac{h_d}{H}.$$

(Example 3)

27. Dynamic Depression Head. A significant feature of the pressure diagrams, Figs. 13-15, is that each of them suggests

that the point of minimum absolute pressure anywhere in the impeller does not occur at the blade inlet. On the contrary, as we follow the liquid as it traverses the *back* of a blade, we notice that there is a distinct fall in pressure-head before the curve begins to sweep rapidly upwards. These areas of depression, in which the pressure is *less* than it is just before the liquid enters the impeller passages, are indicated by horizontal hatching in the diagrams. In this book the maximum depression, i.e., the difference between the head at entry, and the minimum head anywhere on the blade surface, will be termed the *dynamic depression head*, Fig. 15 (i). It will be denoted by h_{dd}.

It seems likely that there will be some relationship between the dynamic depression head and the average differential pressure-head h_d; if h_d has a high value, quite probably h_{dd} will also have a proportionately high value. Since we have already admitted that for all practical purposes it is not possible to plot the true pressure distribution diagram, the only way of assessing the actual value of h_{dd} will evidently be an indirect one. The need for arriving at some sort of estimate becomes pressing when questions of maximum permissible suction lift have to be settled, Chapter XVI.

CHAPTER III

MIXED-FLOW AND AXIAL-FLOW ROTORS

28. Three-dimensional Flow. If the term *mixed-flow* provokes the question: "Mixed with what?" the answer is: mixed with an *axial component.* All the rotors now to be studied differ from those hitherto examined in that the liquid, while flowing through the wheel passages, invariably has an axial velocity component in addition to its radial and tangential components. Already we are familiar with such an axial motion before the liquid comes within the zone of influence of the blades: if we look at any of the relevant diagrams illustrating Chapters I and II, it is evident that the liquid must flow end-wise in order to get into the impeller at all. But we have taken it for granted that the liquid turns through a right angle and begins its purely radial motion before it is ready to receive energy. Already, though, Fig. 16 (iv) and the accompanying explanation, § 25, have shown that such an assumption might not always be justified. Now we no longer pretend to uphold it: it is specifically understood in this chapter that whatever else the liquid is doing—whether it is moving away from the axis or around the axis of the rotor—it must also be moving *parallel* with the axis.

Fig. 18 shows how easily a distortion or "dishing" of a normal radial-flow impeller may introduce the new velocity component. At (i) there is pure radial flow; at (ii) and (iii) we see in section and in perspective the same wheel after it has been pushed over sideways. Although the liquid at a selected point still has its original radial velocity component

Y_r and tangential component V, it now has in addition an axial component that can be designated Y_a.

29. Directive Surfaces. Unless a suitable routine can be framed for dealing with these new complexities, they threaten to grow beyond the range of the simple treatment appropriate for this book. We must contrive some orderly system for bringing under analytic control a multitude of liquid elements all moving very rapidly through the rotor, each element having its own particular radial, tangential, and axial velocity components.

The first step is to assume that in their progress through the rotor passages the elements are guided not only by the metallic walls of the rotor but also by imaginary intermediate surfaces. We may term the metallic walls *control surfaces* and the imaginary partitions *directive surfaces*. Examples as found in a radial-flow impeller are illustrated in Fig. 18 (i). We do not admit that the liquid has the right to wander at will anywhere within the impeller; on the contrary, we regard the elements as being constrained by the imaginary disc-like partitions to keep each in a given radial plane. Just as white lines on the surface of a highway mark out traffic-lanes for road vehicles, so do the control surfaces and the directive surfaces keep each liquid element within its own appointed passage. Just as there may be "slow" and "fast" traffic lanes, so also we shall find it convenient to imagine slow and fast passages for the liquid.

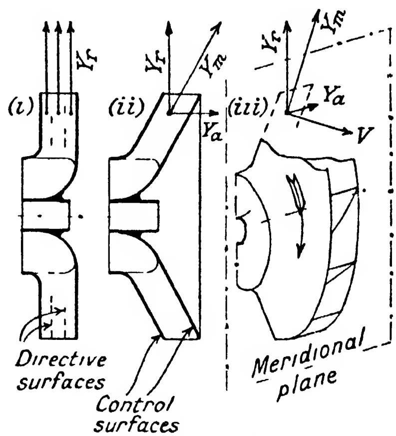


FIG. 18 Comparison between radial-flow rotor (i), and mixed-flow rotor (ii), (iii)

Some possible shapes of directive surface are sketched in perspective in Fig. 19. We begin at (i) with the original disc-like plane surface associated with radial flow, Fig. 18 (i). Next there is the conical surface (ii) which results from dishing the flat surface (i), as in Fig. 18 (ii). It thus becomes clear that

each directive surface must necessarily be a *surface of revolution* ; it can be generated or swept out by causing a line lying in the plane of the rotor axis to rotate about that axis. If the generating line is a straight one, then according to its inclination it will strike out either (i), a disc, or (ii) a cone, or (iv), a cylinder, Fig. 19 ; while a curved generating line will produce a trumpet-shaped surface (iii).

Radial-flow rotors, i.e. centrifugal pump impellers, are represented by Fig. 19 (i) ; *mixed-flow* rotors are represented by Fig. 19 (ii) and (iii) ; *axial-flow* rotors are represented by Fig. 19 (iv).

30. Types of Mixed-flow Rotors. The nomenclature of mixed-flow rotors is still rather indeterminate, but at least the

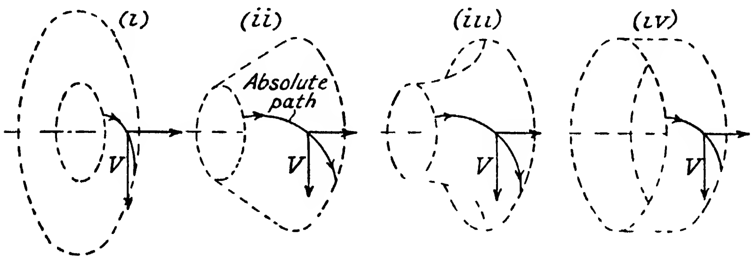


FIG. 19.—Types of directive surfaces. On each surface the absolute path of the liquid is shown.

terms now to be proposed will consistently be used throughout the book.

(*a*) *Diagonal-flow Rotor.* As shown in Fig. 20 (*a*), the blades project from a central conical boss ; both the control surfaces and the directive surfaces are cones, as in Fig. 19 (ii).

(*b*) *Screw Rotor.* Here in diagram (*b*) the surfaces are generated by curved lines, as in Fig. 19 (iii).

(*c*) *Mixed-flow Centrifugal Pump Rotor.* Diagram (*c*) shows that the generating line AOB is still more curved, with the result that by the time the liquid leaves the wheel its axial velocity component has dropped almost to nil.

A sharp distinction between types (*a*) and (*b*), and type (*c*), is that in the first two the rotor blades are secured at one end or side only ; the outer control surface is the *stationary* surface of the pump casing. The unsupported edges of the blades work with a small running clearance against this surface. On

the other hand, type (*c*) in Fig. 20 still has two side plates or shrouds as in a radial-flow impeller; but now one of the shrouds is very much "flared" or opened out in order to provide a sufficiently great inlet area. It is on this account that it is convenient to recognise a well-established convention and to admit that a pump embodying such a rotor shall be termed a centrifugal pump.

Whatever the shape of the wheel, we have to think of the imaginary directive surfaces as being "nested" one within the other like an assembly of plant-pots of assorted sizes.

31. Meridional Plane: Meridional Velocity Component. (*) On comparing the orthographic projection of directive surfaces, Figs. 18 (i) and 20 (*c*), with the perspective views, Fig. 19, it becomes clear that the intersection of any one of the imaginary surfaces with the plane of the drawing or page will mark out the original generating line. This plane—the diametral plane which includes the rotor axis—is termed the *meridional plane.* The reason is as follows: if the generating line were a semicircle with its centre lying on the rotor axis, it would sweep out a spherical surface; and if we compare this surface with a very much larger one—the earth's surface—we recognise that the intersection of the diametral plane with the (nearly) spherical earth's surface would represent a *meridian of longitude.*

In another way also this analogy helps in describing motion over a directive surface. Motion over the earth's surface can be resolved into two components: a north-south component along a meridian of longitude, and an east-west component along a parallel of latitude. The second of these is a tangential component; the first can very properly be named the *meridional* component. Thus the rotodynamic directive surface, Fig. 19 (i), suggests a polar region of the terrestrial surface, while Fig. 19 (iv) corresponds to an equatorial belt.

The separation into two components is only imperfectly indicated in Fig. 19. On each of the directive surfaces there is shown the track or absolute path of a liquid element, and at a selected point of each track a tangential vector shows the tangential velocity component and a horizontal arrow suggests how the meridional vector would appear as foreshortened by the needs of perspective.

32. Analysis of Flow. The meridional velocity Y_m can itself be resolved into a radial component Y_r and an axial component Y_a, Fig. 18 (iii); these together give a measure of *how much* liquid is flowing through the rotor. But we are as much concerned as ever with the tangential component V which helps to tell us how much *energy* is being given to the liquid, § 11. In shaping the blade surfaces, it is still essential to plot velocity diagrams as in Fig. 8. These were two-dimensional diagrams. But even now, although we have a three-dimensional problem on our hands, a two-dimensional diagram is practicable if we plot it on a *suitable plane tangential to the appropriate directive surface.*

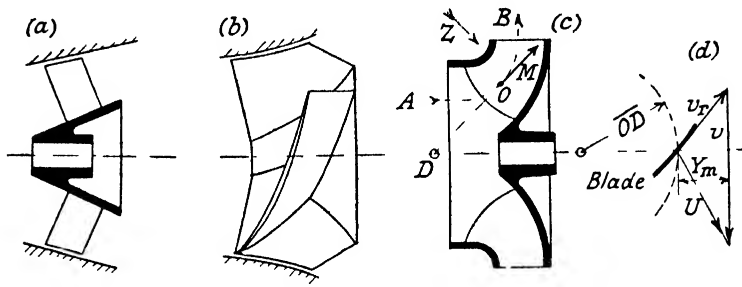


FIG. 20.—Examples of mixed flow rotors.

Suppose we intend to study conditions at the point O of the centrifugal pump rotor shown in Fig. 20 (c). Touching the directive surface AOB at this point is a plane DO perpendicular to the meridional plane (the plane of the paper), and intersected by the rotor axis at D. In diagram (c) the vector OM truly represents the meridional velocity; but it *also* represents an end view of the velocity diagram we are now seeking. If viewed from the direction of the arrow Z, viz. normally to the new plane of projection OD, then the velocity diagram would be seen to have the true shape of Fig. 20 (d). The diagram shows that at the selected point O: the blade surface is moving with tangential velocity v; the liquid has an absolute velocity U; the liquid has a velocity relative to the blade surface of v_r. The whirl or tangential velocity component V could also be sealed off. In order to draw the corresponding local blade shape, it would be permissible to assume that at least for a short distance the directive surface was conical, struck out by

a generating line DO. When unrolled or developed, this surface would have the flat circular form of Fig. 20 (d), and on this surface a short length of the ideal blade could be sketched, parallel with the relative velocity component v_r.

33. Ideal Distribution of Meridional Velocity. Now there arises the question : how in fact are we to compute the value of the meridional velocity at any selected point in the rotor ? In the case of pure radial flow, § 8, we had to depend wholly upon a knowledge of the discharge q flowing through the rotor, and of the cross-sectional area $2\pi rb$ of the annular waterway. We took it for granted that the flow velocity $q/2\pi rb$ was uniform at all points at a given radius, viz. that in each of the annular traffic-lanes of Fig. 18 (i) the liquid elements were moving outwards at the same speed. Such an assumption concerning the rotor of Fig. 20 (c) certainly could not be justified ; already it has been ruled out for a much less developed example of three-dimensional flow, Fig. 16 (iv).

A preliminary impression of how an ideal liquid will distribute itself can be formed by neglecting friction and the disturbing effect of impressed whirl components ; we may study the flow before the blades are inserted, as in § 7. Referring to the mixed-flow centrifugal pump rotor shown in Fig. 21 (i), the generating lines for the various directive surfaces are sketched in so as to fulfil a certain condition : and this condition concerns the relation between the generating lines and the transverse lines ee that intersect them at right angles. The two sets of lines together form a grid or network ; and if the mesh were sufficiently fine we can see that each mesh would have the shape of a tiny rectangle. If dx is the length of the tiny rectangle measured along the generating line, dy is its transverse dimension, and r its mean distance from the rotor axis, then the stipulated condition is that the ratio dx/dy is directly proportional to the radius r.

Naturally a fairly lengthy process of trial and error will be necessary before this requirement is fulfilled at all points. But in the end the completed diagram, Fig. 21 (i), will enable us to say that the total flow of liquid through the rotor will be *equally divided* between the annular spaces delimited by the directive surfaces. Supposing, that is, that the metallic control surfaces and the imaginary directive surfaces have partitioned

off the rotor volume into n spaces, then the flow through each of them will be q/n.

The estimation of meridional velocity at any point is now an easy matter. At the selected point at radius r' we measure the transverse width b' of the traffic lane, and compute the value $Y_m = \frac{q}{n(2\pi r'b')}$, Fig. 21 (i). In this way it may be shown that the "fast" traffic-way o o, is near the outer shroud of the rotor, while the "slow" traffic-way i— i is near the inner shroud.

34. Blade Shapes for Mixed-flow Rotor. To determine the blade form for the rotor shown in Fig. 21 it is necessary to carry out at a number of points the construction for a

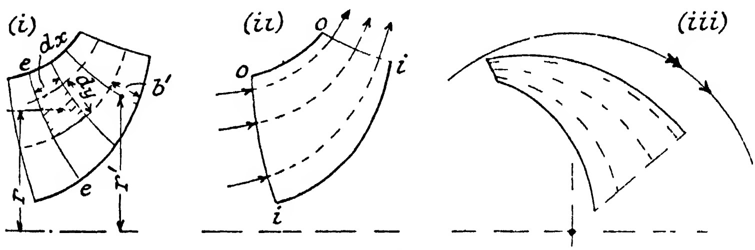


FIG. 21.—Method of establishing blade form for mixed flow centrifugal pump rotor.

specimen point that was described in § 32. For example, we might choose three points on the inlet edge of the blades, and three points on the outlet edge, these representing respectively the points of inlet and exit of liquid elements traversing the middle of each traffic-way, Fig. 21 (ii). We know, or we can readily compute by methods already explained, the values for each point of the meridional velocity Y_m and the peripheral blade velocity v, § 13. While passing through the rotor, all the elements must receive identical and known energy increments $E = V_2v_2/g$, from which corresponding values of tangential components V_2 can be extracted. By measurement of the six resulting velocity triangles, as in Fig. 8, the mean inlet and outlet blade angles appropriate for the three traffic-ways may be set off, and each pair of angles connected by its own smooth curve, Fig. 21 (iii). The complete blade surface can finally be fitted over these guiding lines much as the skin of a

ship is fitted over the curved ribs. It is to be noted that while Fig. 21 (iii) is intended to show a true axial view of the impeller blade, Fig. 21 (ii) only represents a projection of the blade *on to the meridional plane*, § 99.

When the rotor actually gets to work with real liquid flowing through it, there will only be the actual metallic surfaces of shrouds, casing, and blades to guide the liquid. If, as explained in Chapter II, the liquid is not too submissive even to such positive control, how much notice is it likely to take of purely imaginary directive surfaces ? It is one thing to mark out traffic lanes on a highway ; it is quite another to make each driver keep within his allotted lane. So in regard to the liquid elements we cannot promise more than this : that at least we have given them no incentive to wander or " weave " out of their destined path. Since the blades have been designed to transfer added energy at a uniform rate, common to all the traffic ways, the liquid elements have nothing to gain by attempting transverse explorations along the lines *ee*. In any event we must not forget the provisional nature of the basic flow-net, Fig. 21 (i), nor the fact that it will inevitably suffer distortion as soon as the blades begin to impress whirl components on the liquid.

35. Axial-flow Rotors. When the flow through the rotor is controlled (or assumed to be controlled) by cylindrical control and directive surfaces, Fig. 19 (iv), the complexities of design appear to be mitigated. Radial velocity components are now wholly absent ; the meridional velocity becomes identical with the axial component ; and ideally the axial component is uniform at all points in the rotor. Henceforward the designation *velocity of flow* will serve equally well either for Y_m or for Y_a, and its value can at once be computed from the equation of continuity, § 36.

In Fig. 22 a 4-bladed axial-flow rotor is shown working within a cylindrical fixed casing, lifting liquid through a vertical height H. The two specimen cylindrical directive surfaces sketched in perspective relate respectively to a point near the rim and a point near the boss. In accordance with our belief that no pre-rotation exists, § 17, the absolute velocity vectors at inlet, having the magnitude $Y_a = U_1$, are drawn vertically ; but as a result of the tangential acceleration that the blades

have impressed on it, the liquid when leaving the rotor has a helical or corkscrew motion. It is to be noticed that near the boss the absolute outlet velocity vector U_{2a} is much more sharply inclined than it is near the rim, U_{2b}.

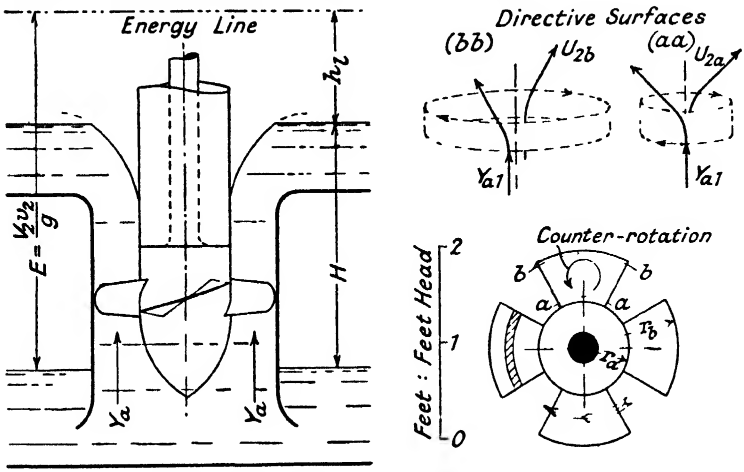


FIG. 22.- Elementary type of axial-flow pump.

36. Blade Form for Axial-flow Rotor.

Unrolling or developing the directive surfaces of Fig. 22 yields the flat strips seen in Fig. 23 (i). On these the velocity diagrams and blade forms may be constructed thus :—

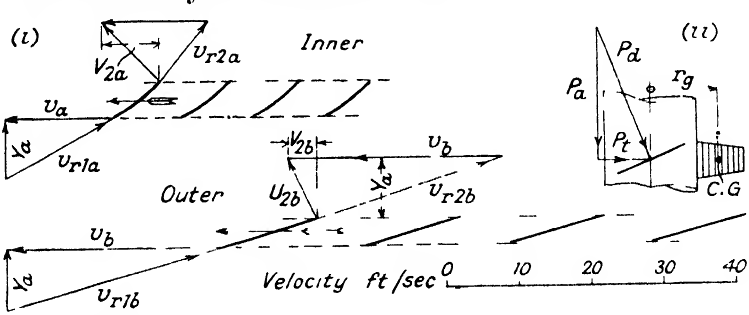


FIG. 23.— Velocity diagrams, and developed directive surfaces, for axial-flow rotor.

Assuming that r_b = outer radius of blade = radius of outer directive surface,

r_a = inner radius = radius of inner directive surface,

then $\quad v_b =$ outer peripheral blade velocity $= 2\pi r_b N/60$,
$\quad v_a =$ inner peripheral blade velocity $= 2\pi r_a N/60$,
also if $\quad E =$ energy in foot-pounds per pound to be transferred to liquid,
then $\quad V_b =$ outer exit whirl component $= \dfrac{gE}{v_b}$,
$\quad V_a =$ inner exit whirl component $= \dfrac{gE}{v_a}$,
and $\quad Y_a =$ velocity of flow $=$ axial component $= q/\pi(r_b^2 - r_a^2)$.

All information is now ready for plotting the inlet and outlet velocity diagrams, and the blade profiles are drawn parallel, at inlet and outlet, to the respective relative velocity vectors, Fig. 23 (i).

If we had been interested in "datum" blades instead of in "working" blades, §§ 9, 10, such that no energy was transmitted to the liquid, then the developed blade profiles, Fig. 23 (i), would have been straight lines, and the blades would have had a true helicoidal form.

37. Efficiency and Axial Thrust. Still accepting the ideal conditions that applied throughout Chapter I, we can now compute the vertical height through which the rotor has raised the liquid elements. From equation (1-3), § 12, this distance is *not uniform* for all the elements, because the absolute velocity of rejection, U_2, is not uniform. As just shown in § 36, since the blade peripheral velocity varies directly as the radius, then the tangential velocity at exit must vary *inversely* as the radius. In other words, the velocity distribution in the space above the rotor fulfils the law of *free vortex flow*, and that is why the ideal free liquid surface is correspondingly curved, Fig. 22. This ideal curvature is identical with the curvature assumed by water which has formed itself into a vortex or whirlpool when escaping through a circular hole in the flat bottom of a tank.

If we assume that the liquid finally flows away at the level it attains at the outer rotor periphery, Fig. 22, then the efficiency of the apparatus is not very high; an amount of energy represented by h_l in the diagram is wasted in eddying.

In regard to the tangential component of the total load on

each blade, the rough method described in § 26 is equally applicable here. It is evident from Fig. 23 (ii) that now the total load P_d as well as the axial component P_a are both much greater than the tangential component itself, P_t. The value of this axial component, P_a, has an important influence on the design of the pump, for a thrust bearing must be provided capable of taking the quite considerable gross thrust on all the four blades of the rotor.

38. Aerofoil Type of Axial-flow Blading. An interesting alternative treatment of blade thrust depends upon the view that the blade may be likened to an *aerofoil*, i.e. the wing of an aeroplane or the blade of an airscrew. We are the more disposed to follow up this comparison because of the disparaging remarks about a solitary blade that were made in § 17. Are not the axial-flow blades illustrated in Figs. 22 and 23 spaced so far apart that they are themselves virtually solitary ? At any rate they do not seem very well disposed for giving mutual support one to the other. So it is reassuring to be offered the chance of comparing the propeller of a pump with the propeller of an aircraft ; if the widely-pitched blades of the airscrew do not seem to handicap it, then we may be less dubious about the blades of the axial-flow rotor.

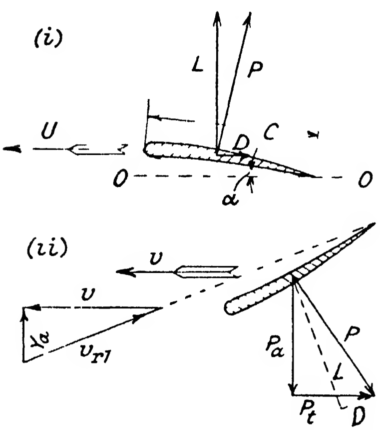


FIG. 24.—Forces on aerofoil and on pump blade.

In truth a solitary element of the proper shape is by no means to be derided : it can give a very good account of itself. In Fig. 24 (i) such an aerofoil is represented moving with velocity U through a mass of stationary liquid. The *chord length* is measured as shown, and has the value c ; the span or length of the aerofoil in a direction at right angles to the paper is b ; while the *angle of attack*, or angle between the chord and the direction of motion, is denoted by α.

The total force or reaction of the liquid on the moving aerofoil, of intensity P, can be resolved into components parallel with the direction of motion oo and normal to this direction. They are denominated respectively :—

Drag, D, is the parallel component,
Lift, L, is the normal component, Fig. 24 (i).

39. Forces on Blade Element. The Drag and the Lift on the aerofoil may be evaluated from the expressions :—

$$\left.\begin{aligned} \text{Drag } D &= C_D \,.\, w \,.\, b \,.\, c \,.\, \frac{U^2}{2g} \\ \text{Lift } L &= C_L \,.\, w \,.\, b \,.\, c \,.\, \frac{U^2}{2g} \end{aligned}\right\} \quad . \quad . \quad (3\text{-}1)$$

where C_D is the *drag coefficient*, and C_L is the *lift coefficient*. Their values depend chiefly upon the shape of the aerofoil and upon the angle of attack, α.

The *work done per second*, or energy expended in unit time in forcing the blade through the liquid against the resistance of the drag component, will manifestly be expressed by $U \,.\, D$.

The lower diagram, (ii) in Fig. 24, shows the aerofoil turned over until it has taken up the posture of a propeller-pump blade. We may now regard this diagram as a side view of the blade element that is distinguished by hatching in Fig. 22; and we observe that the position has been adjusted so that the original datum line oo is now parallel with the inlet relative velocity v_{r1} as obtained from the appropriate velocity diagram. Since the vector v_{r1} in diagram (ii) corresponds with the vector U in diagram (i), we may expect that the formulæ (3-1) above will still hold good, provided that v_{r1} is substituted for U. But the expression ($v_{r1} \,.\, D$) now only tells us what is the energy per second dissipated in *overcoming the drag;* to obtain the value of the total energy absorbed in forcing the blade element *through the liquid* we must use the expression $v \,.\, P_t$, where v is the peripheral velocity of the element and P_t is the *tangential component* of the total thrust P, Fig. 24 (ii).

If, therefore, we feed into the blade element a total amount of energy per second $v \,.\, P_t$, and we waste an amount $v_{r1} \,.\, D$,

then presumably the net energy received by the liquid is $(v \cdot P_t - v_{r1} \cdot D)$. Of this net quantity, some has been used up in increasing the velocity energy of the liquid from $\frac{Y_a^2}{2g}$ to $\frac{U_2^2}{2g}$, Fig. 23 (i), and so we may say that the balance represents the *increase of pressure-head* impressed on the liquid. The argument is a good example of the general policy laid down in § 3 (iv): we first make sure of supplying energy to the liquid, and only afterwards do we ask what happens to the energy.

To compute the gross energy that must be fed into the rotor, it will evidently be necessary to repeat the process for other blade elements, and to sum up the contributions that each part of every blade can offer. Similarly the total axial thrust, that in § 37 was computed by an approximate method, might now be estimated by summing the various values of the individual thrusts P_a on each element, Fig. 24 (ii).

40. Comparisons between Radial-flow and Axial-flow Rotors. Although the pumps so far studied are as yet in an immature and undeveloped form, it may already be profitable to compare their respective behaviour. The machines shown in Figs. 8 and 22 respectively have the following data in common :—

Rotor diameter	2 ft.
Peripheral speed at outer rim	30 ft./sec.
Velocity of flow	8 ft./sec.

Ideally they would give the following performances :—

	Centrifugal (Radial-flow) Pump	Propeller (Axial flow) Pump.
Ideal lift (feet) . . .	11·4	2·5
Discharge (cub. ft./sec.) .	5·0	18·9
Number of blades . .	8	4
Blade loading ϵ (§ 26) .	0·73	1·95
End thrust on rotor, lb. .	0	480

While, then, the centrifugal pump seems to be better suited for generating high pressures than the propeller pump,

yet the axial-flow pump can deal with much larger volumes of liquid.

41. Corrections to Simple Theory. In regard to mixed-flow and axial-flow pumps, the theories advanced in this Chapter are on a level with those applied in Chapter I to radial-flow pumps ; they relate only to ideal rotors having an infinitely great number of blades. In Chapter II it was shown that as the blades of an impeller became less and less numerous, other things remaining equal, so also did the tangential velocity impressed on the liquid progressively diminish. If that was true when the blade number was reduced to (say) eight, we cannot expect any mitigation of the process when the rotor only has four blades, as it has in the axial-flow pump shown in Figs. 22 and 23. To form a correct impression of the behaviour of the axial-flow rotor, § 35, or the mixed-flow rotor, § 34, therefore, the respective velocity diagrams will require correction on the lines of the modification illustrated in Fig. 16. The actual whirl component V_n impressed on the liquid elements, which is what ultimately determines the rate at which energy can be fed to the liquid, will invariably be *less* than the ideal velocity as scaled from the velocity diagram applicable to an infinite number of blades.

It is possible that the effect of counter-rotation, § 23, may be slightly different in the axial-flow rotor from what it was in the radial-flow rotor, Fig. 15 (iii). The arrow in the plan view in Fig. 22 suggests that the relative velocity v_{r2b} of the liquid over the blade at the outer radius will be increased, and reference to Fig. 23 (i) shows that this will necessarily reduce the value of the corresponding outlet whirl component V_{2b} ; so at least in this respect the effects are comparable.

Other matters that will require study when real rather than ideal pumps are to be built include :—

(i) It will be vain to expect the simple types of velocity distribution postulated in §§ 33, 35. The metallic control surfaces of the rotor do not merely guide the liquid—they very materially influence its local velocity. If we never find uniform velocity prevailing across a diameter of a long straight circular pipe—if the local velocity near the pipe walls is invariably less than it is near the pipe axis—it is inevitable that some similar

tendency will be manifest in the annular passages of the rotor. Thus in the three traffic-ways depicted in Fig. 18 (i), there will *not* be the uniformity represented by the equal arrows Y_r; on the contrary, we may expect faster motion in the centre one than in the outer ones.

(ii) When we return to reality after studying the ideal single aerofoil, §§ 38, 39, we shall remember that the rotor has multiple blades, and that their influence one upon the other may sensibly modify the original theory applicable to a single aerofoil.

CHAPTER IV

THE RECUPERATOR: THE COMPLETE PUMP

42. The Casing and its Duties. A rotor can only effectively deliver energy to the liquid flowing through it if there are proper means for guiding the liquid into the rotor and for collecting the liquid as it comes out again. Crude forms of casing were indicated in Figs. 4 and 8; but it is by no means sure that the inlet and outlet passages had the best possible shape. At any rate they seemed to be responsible for a considerable waste of energy—a waste that was graphically represented in Fig. 8. The reason for the loss was explained in § 12; the fundamental principle of the rotodynamic pump requires the liquid in the rotor to be accelerated, and unless the resulting velocity energy is carefully guarded the liquid cannot profit by it. The increase in velocity energy is a valuable by-product of the process that we can no longer afford to neglect.

The *recuperator* is that part of the pump casing that lies between the rotor outlet and the pump outlet or delivery branch; and it is so called because it provides the opportunity for the recuperation of pressure-head—for the conversion of velocity energy into pressure energy. The types of casing so far proposed failed in this object because they were the wrong shape; now our aim is to develop improved shapes. In this way, without any increase whatever in the energy *input* to the pump, we shall increase the useful energy *output*. The rise in pressure-head which it is the purpose of the pump to generate will be carried out in two steps: (i) in the rotor there will be a rise of h_i, (ii) in the recuperator there will be an *additional* gain of h_g.

To be successful, the planning of the recuperation process requires a frame of mind wholly different from what has hitherto

prevailed. While the liquid is actually within the rotor passages we can be as masterful as we like ; but it is useless to be authoritative when once the liquid has escaped into the casing. It is the liquid that has the upper hand now : it has collected its full quota of energy, and only by applying the right treatment can we coax it into yielding up again the additional pressure-energy we are seeking. If we are brusque or inconsiderate, the liquid will show its displeasure in the usual way—it will throw away energy.

43. Principles of Recuperation. (*) The liquid enters the recuperator of a rotodynamic pump with velocity U_2—the absolute velocity of rejection from the rotor—and it leaves the recuperator with a much lower velocity U_3—the velocity at the

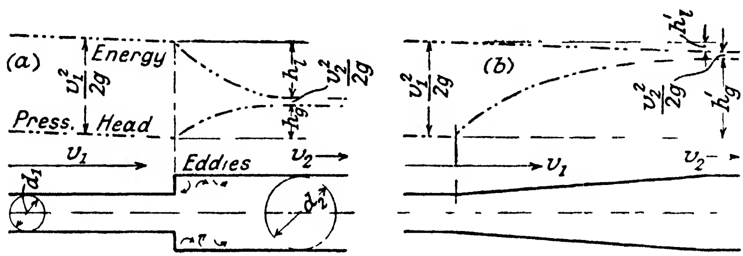


FIG. 25.—Principle of recuperation in circular pipes.

delivery branch. The basic problem is to conduct this retardation as gently as possible. Two general systems are available :—

(i) *Diverging Conical Passage.* Fig. 25 shows alternative methods of coupling a small circular pipe to a larger circular pipe. In the one, (a), the transition is abrupt ; in the other, (b), the transition is gradual. Although the initial and final velocities v_1 and v_2 are the same in both cases, yet the recuperation of pressure-head is very different in amount. The sudden enlargement (a) only yields an increment of pressure-head h_g that is less than half of the original supply of velocity energy, and at least half of this supply, h_l, is wasted in eddying and turbulence. On the other hand, in the gradual enlargement (b) we lose relatively little on the exchange. An expenditure of energy of h_l' yields a gain of pressure-energy of h_g'.

(Example 4)

(ii) *Vortex Chamber.* This is the name given to an annular type of recuperator specially adapted for centrifugal pumps,

Fig. 26; it consists of two flat parallel plates set the same distance apart as the width of the impeller mouth. Liquid elements enter with the flow and whirl components derived from the impeller, and thereafter the motion is governed by the conditions of free vortex flow, viz. uniformity of energy. In consequence the elements follow a spiral path as shown in the diagram; the absolute velocity progressively declines as indicated by the vectors, and the corresponding *ideal* pressure-recuperation is plotted in the upper view. **(Example 5)**

In practice the vortex chamber or whirlpool chamber is not nearly as satisfactory as this: the recuperation is relatively imperfect, resulting in quite perceptible energy losses. Taking advantage of its freedom to go where it likes, the liquid seems to grow confused and uncertain, and the flow is not uniform around the circumference of the chamber.

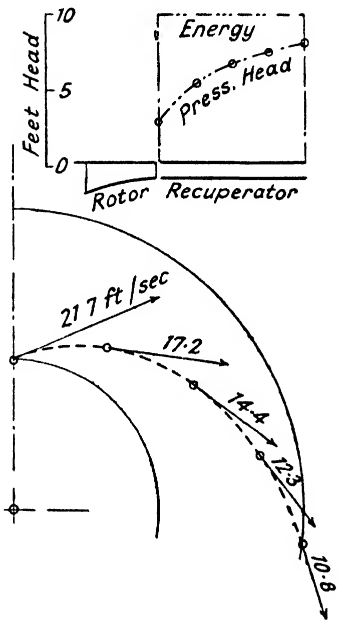


FIG. 26.—Ideal changes of velocity and pressure-head in vortex chamber.

Recuperators for Centrifugal Pumps

44. Diffuser or Guide-ring.

If the vortex chamber is fitted with fixed vanes adapted to guide the liquid positively along its spiral path, Fig. 27, then the resulting recuperator is known as a *diffuser* or *guide-ring* or *diffuser-ring*. It thus has affinities with both of the two systems described in § 43; we can regard it either as a modified vortex chamber, or else as a series of curved divergent passages, Fig. 25 (*b*), disposed around the impeller rim. In Fig. 27 the diffuser is seen applied to the original pumping outfit of Fig. 8, and it discharges into an annular casing of increasing cross-section which ultimately conducts the liquid to the pump delivery branch.

Its effect on performance is most marked. As compared with the original casing which wasted a head of 6·2 ft. and

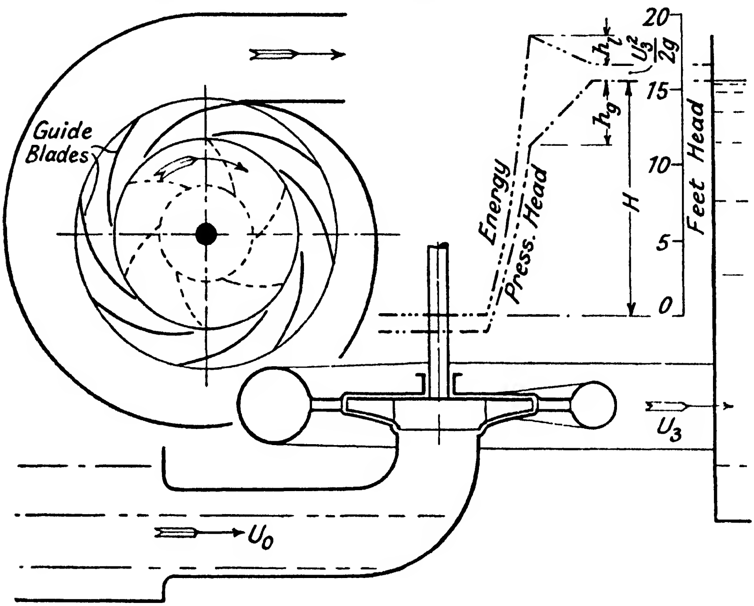


FIG. 27.—Diagram showing how performance of simple pump in Fig. 8 is improved by addition of diffuser-ring.

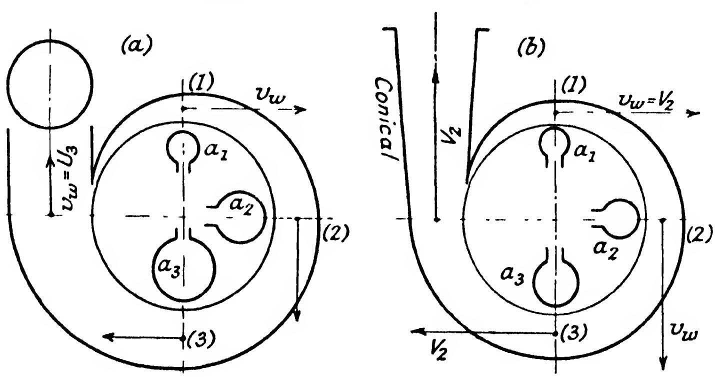


FIG. 28.—Types of volute casings (i), variable-velocity (ii), constant velocity.

yielded no pressure-increase in return, we see that now the energy loss has been brought down to 2 ft. while the head regain h_g is 4·2 ft. Thus the pump now raises the liquid

through a total height of 15·6 ft. instead of 11·4 ft. as formerly. Yet this additional useful lift is a *free gift*; it involves no extra power input to the pump-shaft.

45. Volute Casing. (*) (*a*) *Variable-velocity Volute.* If the diffuser-ring, § 44, may be likened to a series of diverging passages, might it not be permissible to use a single diverging passage wrapped round the entire circumference of the rotor? The resulting casing would then have the shape suggested in Fig. 28 (*a*). It is not difficult to proportion the successive cross-sections of the volute in such a way as to encourage the desired gradual retardation of the liquid, from velocity U_2 to velocity U_3. Remembering that the liquid escapes uniformly all round the impeller circumference, we observe that at point (1), for example, the volute has to accommodate one-quarter of the total flow, at point (2), one-half of the flow, and (3), three-quarters of the flow. Thus area $a_1 = \dfrac{q}{4v_{w_1}}$, etc., where v_{w_1} represents the mean tangential velocity of the liquid in the casing.

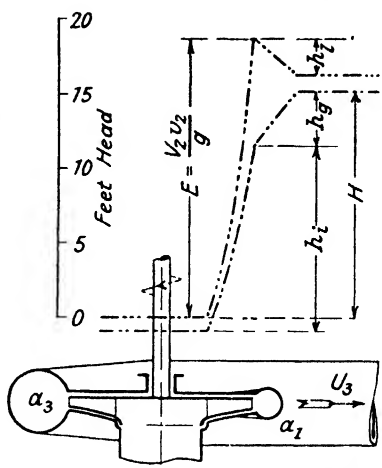


FIG. 29.—Energy and pressure-head lines for pump with volute.

(*b*) *Constant-velocity Volute.* In the alternative form of volute, Fig. 28 (*b*), the two functions of the casing are separated: a circumferential part merely collects the liquid without attempting to modify its velocity, while a conical outlet branch identical with the tapered pipe enlargement, Fig. 25 (*b*), carries out the pressure recuperation in the most effective manner. The cross-sectional areas at the points (1), (2), etc. are therefore contrived so that everywhere the mean liquid velocity v_w is the same as the outlet whirl component V_2.

Either of the casings can be relied upon to improve the pump performance in the manner indicated in Fig. 29, which may be compared with Fig. 8 and Fig. 27.

46. Comparative Comments. (*) The best way to design the most efficient recuperator is not necessarily to think only of the maximum amount of pressure-head we can recover ; if we concentrate rather upon keeping down energy losses to the minimum, then automatically that will ensure the greatest measure of head recuperation. As in all similar hydraulic devices, there may be energy losses due to eddying and turbulence, and those due to friction. In the original casing that was condemned as a recuperator, Fig. 8, eddying accounted for the whole of the loss ; this eddying or state of extreme turbulence was caused by the impact between the fast-moving stream issuing from the impeller and the (assumed) nearly stationary liquid in the casing.

In the variable-velocity volute, Fig. 28 (*a*), conditions vary from point to point. At (1) the liquid in the casing is assumed to have nearly the same velocity as the liquid leaving the impeller, so that shock losses are relatively small. On the other hand, the friction loss is high because of the relatively high speed with which the liquid moves over the casing walls. But at point (3) the tendencies are reversed : the relative eddy loss has increased while the friction loss has diminished. There are no shock losses in the constant-velocity volute, Fig. 28 (*b*) ; friction seems to be predominant, for everywhere in the circumferential part of the casing the liquid velocity has the very high value of V_2. Is this indeed true ? Is the velocity uniform over any cross-section of the volute ? This seems to be extremely improbable. Even in a straight pipe we do not expect uniform velocity-distribution ; in a curved pipe we certainly do not get it.

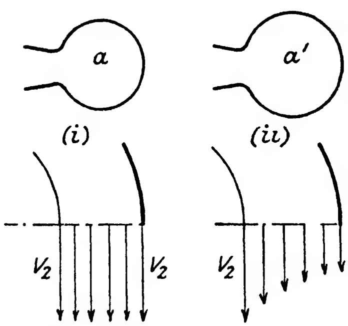


FIG. 30.—Volute cross-sections based on (i), uniform velocity distribution (ii), free vortex flow.

If, then, we make the quite justifiable assumption that the liquid near the inner part of the volute adjoining the impeller is moving faster than the mean filaments in accord with the laws of free vortex-flow, § 43 (ii), then the *mean* velocity

may be less than is indicated in Fig. 28 (*b*). The successive cross-sections can be rather more generous than those shown in the original diagram, with the result that the friction loss in the volute will be materially less than we feared. Fig. 30 (i) shows the cross-section (*a*) of the volute based on the original assumption of uniformly-distributed flow ; Fig. 30 (ii) shows the greater area (*a'*), based on free-vortex flow. **(Example 6)**

In regard to friction losses in the vortex chamber, Fig. 26, it will be noticed that these grow more serious as the flow velocity of the liquid sinks. The smaller the radial velocity component, for a given tangential component, the longer becomes the curved path traced out by liquid elements before they escape from the annular chamber. One of the advantages of the diffuser blades, Fig. 27, is that they can be shaped to divert the liquid quickly to the outside of the casing and in this way shorten the path. Naturally the inlet tip of the blades must be set parallel with the outlet absolute velocity vector U_2 of the liquid leaving the impeller.

Recuperators for Mixed-flow and Axial-flow Pumps

47. Screw Pumps and Half-axial Pumps. Two types of casing are available for the screw rotor illustrated in Fig. 20 (*b*).

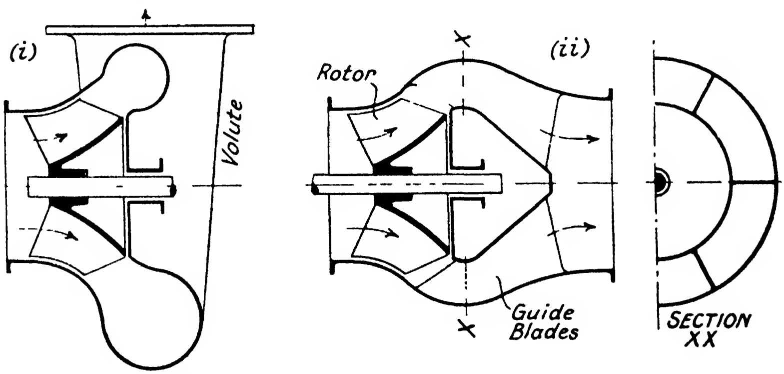


Fig. 31.—Recuperators for (i) screw pump, (ii) half-axial pump.

If the recuperator has the form of a volute, § 45, then the complete machine is entitled a *screw pump*, Fig. 31 (i) ; if the recuperator is in line with the rotor axis, then the title *half-axial pump* is preferred, Fig. 31 (ii). Like so many terms associated

with rotodynamic machinery, these distinctive names are not universally recognised, but they will be used consistently throughout this book.

The only radical difference between the volute shown in Fig. 29 and the one in Fig. 31 arises from the difference in the relative volumes of liquid they have to handle. As is easily visible to the eye, the screw-pump volute has a much wider passage than the centrifugal pump volute has. A minor point is that in Fig. 31 (i) the liquid has a considerable axial velocity component at the moment of entering the casing, whereas no such component is apparent in Fig. 29. A special type of recuperator, Fig. 65, § 101, can effectively deal with this axial component (*).

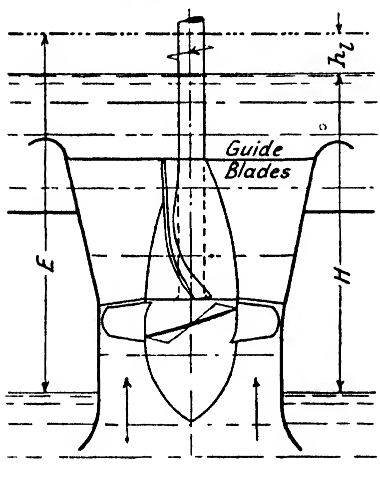


FIG. 32.—Rotor and recuperator for propeller pump.

In the recuperator for the half-axial pump, Fig. 31 (ii), the liquid flows between an outer pear-shaped casing and an inner concentric cone which also houses a bearing for the rotor shaft. Guide-vanes divide this annular space into a number of passages, the inlet edges of the guide-vanes being adapted to the angle at which the liquid leaves the rotor blades. The outlet edges of the guide-vanes are formed so as to deliver the liquid axially into the delivery pipe.

48. Propeller Pump. An axial-flow rotor, § 35, together with its recuperator, may be termed a propeller pump or an axial-flow pump. The recuperator, Fig. 32, combines the principle of the diverging conical pipe, Fig. 25 (*b*), with that of the guide-vanes used in the half-axial pump, Fig. 31 (ii). Not only is the outer casing flared out, but the inner fixed boss is likewise "flared", tapered or stream-lined. The diagram gives an impression of how the inlet angle of the guide-blades varies from point to point along a radius, in harmony with the

changing inclination of the outlet absolute velocity vectors, Figs. 22 and 23 (i). As in the half-axial pump, the guide-blades serve also as webs which tie together the outer casing and the inner fixed boss.

Comparison between Figs. 22 and 32 suggests how effective the recuperator should be. The inequalities of pressure-head explained in § 37 have now been smoothed out, the overall energy loss has in this instance been reduced to 0·4 ft., and the useful lift has risen from 2·5 ft. to 3·3 ft., *without increase of energy input.*

The Complete Pump

49. Some Revised Conceptions. With a knowledge of the main hydraulic requirements that the pump components must fulfil, we should be ready to proceed with the mechanical design of these elements, as explained in Part B of the book. For the moment it may be worth while to notice how the elementary equations that represented pump performance in § 13 are already in need of correction. Thus the expression for head generated (1-4), takes no account of the additional head furnished by the recuperator, and the expression for efficiency (1-6), is likewise inadequate. But even if one were to add to the head generated in the impeller the head regained in the casing, the sum would not yield a figure that would agree with test-bed results ; there remain energy losses in the impeller that must still be evaluated and taken into account.

Although we shall expect to find that the effect of the impeller blade angle on the pump efficiency is less marked than appeared in § 14, yet it seems probable that the dependence of the one on the other has not been wholly suppressed. Selecting, for example, the impeller having a 90° outlet angle, Fig. 10 (*c*), we observe that the liquid leaving the impeller mouth is carrying away with it an amount of velocity energy equivalent to 53 per cent. of the energy originally given to it. The corresponding figure for the 20° impeller (*a*) is only 27 per cent. We admit that the complete pump in which the impellers are mounted will include a recuperating device, but we are quite certain that conversion losses will occur in that device. As it seems likely that this loss will bear a fairly constant ratio to the

quantity of energy to be converted, we conclude that the *overall* energy loss associated with the 90° blade angle will still be more serious than when the angle is reduced to 20°.

As for the general relationship between the rotor and the recuperator, it may be pertinent to refer again to the original basic diagram, Fig. 1 (ii). Although that diagram was never intended to give more than a very broad impression, we now realise that the function of the recuperator does closely correspond to the function of the ramp that converted the velocity energy of the moving solid into potential energy. It is also useful to trace once again the track of a liquid element on its way through the pump. We might piece together in their correct relative positions, for example, the tracks plotted in Figs. 7 and 26. Although they are both curved paths, it is important to remember that while traversing the rotor passage the liquid element is steadily *increasing* its tangential velocity, while in the recuperator the tangential velocity is continually *diminishing*. The *pressure-head* rises rapidly in the rotor, and relatively slowly in the recuperator. The *total energy* increases quickly while the liquid element is in the rotor passage, and perceptibly falls in the recuperator.

In brief, the rotor puts the whirl component into the liquid, and the recuperator takes most of it out again.

CHAPTER V

STANDARDS OF COMPARISON FOR PUMPS

50. Need for Non-dimensional Standards. Rotodynamic pumps can be built in a very great variety of sorts and sizes. The rotor may look like a wheel or it may look like a screw propeller. It may be small enough to go into a coat pocket or it may be so large that only a crane can lift it. The speed in revolutions per minute may be 50 or it may be 10,000. The pressure-difference generated may be 1 lb./sq. in. or 500 lb./sq. in. The liquid may flow through the pump at the rate of a cupful per second or at the rate of 10 tons per second. When designing or choosing a pump, moreover, the engineer has not only to adapt himself to this extremely wide range of performance: the performance itself may be described by a variety of combinations of units. The pressure-difference may be stated in terms of pounds per square inch, feet head, metres head, atmospheres, kilograms per square centimetre, and so on. The flow may be expressed in litres per second, gallons per minute, millions of gallons per day, U.S. gallons per hour, cubic feet or cubic metres per second, etc., etc. Finally there are numerous ways of specifying the density, viscosity, and temperature of the liquid to be pumped. So it seems as though the task of effectively comparing one pump with another might be formidable.

51. Guiding Rules. Nevertheless the consistent basic principle that underlies the operation of all types of rotodynamic pumps should encourage the search for a solution. From other branches of Hydraulics and Fluid Mechanics, too, we

might find useful guidance. These suggest two general rules: (i) To discard if necessary the units that serve for commercial and industrial purposes and replace them by a mathematically consistent system, e.g. feet head, cubic feet per second; or decimetres head, litres (cubic decimetres) per second; or metres head, cubic metres per second, and so on. (ii) To escape whenever possible from the tyranny of irrational units by using non-dimensional *ratios* to describe the pump and its performance. The resulting group of pure numbers will then be independent of the system of units preferred by the pump user or the pump manufacturer, and if possible they will be unaffected even by the size of the pump.

Such a quest will lead to a highly desirable goal. Just as a golfer uses a numeral to describe the shape of a golf-club, so an engineer can use a number to specify the proportions of a pump rotor. The experienced player knows when he must take a number 3 club and when he ought to try a number 8 club. Similarly the engineer versed in non-dimensional terms will know that for a boiler-feed pump he should choose a rotor described by the number 60, whereas for a low-lift drainage pump he would do better to keep within the range of numbers 300 to 800.

The variables primarily involved, and that must first be converted if necessary into the correct units, are usually :—

Speed, N, revs. per min.
Head, H.
Discharge, Q.
Mean diameter of rotor at outlet, d_2.
Mean outer peripheral velocity, or rim speed, of rotor, v_2.
Width of centrifugal pump impeller at outer diameter, b_2.
Velocity of flow of liquid, Y.

52. Width Ratio, Speed Ratio, Flow Ratio.* These simple expressions reveal the relationship between *two* variables.

(*a*) *Width Ratio.* This is the ratio :—

$$\frac{\text{Width of impeller mouth}}{\text{Diameter of impeller}} = \frac{b_2}{d_2}.$$

It is denoted by the symbol λ.

(*b*) *Speed Ratio.* Although we cannot directly compare a head with a velocity, we can compare a given velocity with the

velocity which the *given head would ideally generate.* An object falling freely *in vacuo* through a height H would acquire a velocity $\sqrt{2gH}$; similarly a liquid escaping from an orifice under a head H would attain an ideal velocity of $\sqrt{2gH}$. This value $\sqrt{2gH}$ is sometimes termed the *spouting velocity;* it serves as a useful standard of comparison.

$$\text{The ratio} \qquad \frac{\text{Peripheral velocity}}{\text{Spouting velocity}} = \frac{v_2}{\sqrt{2gH}}$$

is termed the *speed ratio* and is denoted by ϕ.

(c) *Flow Ratio.* This is the name given to the ratio :—

$$\frac{\text{Velocity of flow}}{\text{Spouting velocity}} = \frac{Y}{\sqrt{2gH}}.$$

It is given the symbol ψ.

53. Conditions for Unchanged Efficiency. Advancing now to the more general problem of comparing (say) pump (A) with pump (*), and using the subscripts $_{(a)}$ and $_{(b)}$ to distinguish them, we can make little progress so long as the respective performances are expressed in their original forms: N_a, H_a, and Q_a; and N_b, H_b, and Q_b. At least in one respect the two machines must be brought to a common basis. We may ask, for instance, how the pumps would behave if they both worked under the *same* head or if they both revolved at the *same* speed. Such a proposal at once raises the further question, is it fair to the pumps to ask them to work under conditions for which they were never designed?

Such solicitude for the pump's rights and privileges is wholly commendable. But the machine will have no cause for complaint. Nothing unreasonable will be demanded of it. We say to the pump, in effect: when working under the new head you need not necessarily run at the old speed: you can choose your own speed. What speed is the pump likely to prefer? Surely the one which shows it to be the best advantage: the speed which still enables it to run with undiminished efficiency if this is possible. By studying the original and the modified outlet velocity diagrams, Figs. 8 and 33, we find that the necessary conditions are :—

(i) Since it is still essential that the velocity diagrams shall conform to the blade angles of the rotor, then the triangles

must be *geometrically similar.* Using the subscript $_x$ to denote the new or standard conditions, we can write :—

$$\frac{v_2}{V_2} = \frac{v_x}{V_x}, \text{ and } \frac{v_2}{Y_2} = \frac{v_x}{Y_x}$$

(ii) Since the pump efficiencies are to remain unaltered then it follows from § 13 that $\frac{gH}{V_2 v_2} = \frac{gH_x}{V_x v_x}$. From (i) above, this can be put in the form $\frac{gH}{V_2 v_2} = \frac{gH_x}{\frac{v_x}{v_2} \cdot V_2 \cdot v_x}$, which reduces to the form $\frac{H}{v_2^2} = \frac{H_x}{v_x^2}$.

(iii) Since, in a given impeller, $Q \propto Y$, $Y \propto v$, and $N \propto v$, we see that $Q \propto N$.

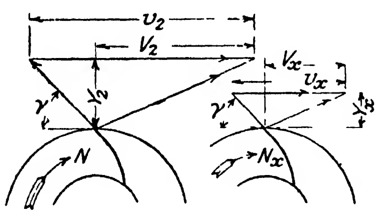


FIG. 33.—Effect on velocity diagrams of speed variation.

(iv) Since power, P, depends upon $W . H - Qw$ H, § 13, then from (ii) and (iii) it follows that $P \propto N^3$.

In brief, the conditions for unaltered *ideal* efficiency under changing head are that the *speed ratio* and the *flow ratio*, § 52, remain unchanged. This implies that :-

The *discharge* will vary directly as the *speed*, N.
The *head* will vary as the *square* of the speed, N^2.
The *power* will vary as the *cube* of the speed, N^3.

Although these ideal relationships will serve for general purposes of comparison, the true relationships are slightly modified by the effect of friction, etc., § 220.

54. Geometrically-similar Rotors. The conceptions of standard head and standard speed only take us part of the way along the road to complete equality of opportunity. Even if a small pump and a large pump were both to be tested against the same standard head, the small pump would still feel at a disadvantage. The next step is clear: we must compare pumps of standard *diameter:* we must study the behaviour not of the actual rotor, but of an imaginary larger or smaller

one of standard size. Naturally the actual rotor and the standard one must be identical in shape and proportions: they must have the same blade angles, the same width ratio, § 52, and in short they must be built from the same working drawings except for the scale. The subscript $_y$ will distinguish the performance of the standard rotor.

In Fig. 34 the two rotors are compared, running at rotational speeds so adjusted that the peripheral velocities v_x and v_y are identical. This implies, § 53, that the velocity diagrams are not merely similar but actually *identical*, from which it follows that comparable velocities are identical throughout, and that the *heads* are also identical; the stipulations of unaltered efficiency, speed ratio, and flow ratio hold good here just as they did in § 53.

The ratio of the *speeds* will be :—

$$\frac{N_x}{N_y} = \frac{\frac{60v_x}{\pi d_2}}{\frac{60v_y}{\pi d_y}} = \frac{d_y}{d_2},$$

viz. the speeds are inversely proportional to the diameters.

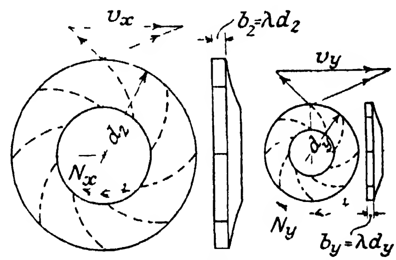


FIG. 34. Rotors of different sizes generating identical heads.

The ratio of the *discharges* will therefore be :—

$$\frac{q_x}{q_y} = \frac{\pi d_2 \lambda d_2 \, . \, Y_x}{\pi d_y \lambda d_y \, . \, Y_y} = \frac{d_2^2}{d_y^2},$$

viz. the discharges are proportional to the *square* of the diameters.

It will be observed that idealised rotors have again been chosen; again it is permissible *for present purposes* to neglect the thickness of the blades, the roughness of the passages, etc., because these effects are examined in detail in § 225.

55. Reduction to Standard Conditions. We now have the means of reducing the performance of any pump to standard conditions—conditions which at last will permit a fair and reasonable comparison between even the most diverse types of machines. Two systems are available: (i) performance under unit conditions, (ii) performance under specific conditions. In

this paragraph which deals with *unit* conditions, we compute the ideal performance of a pump geometrically similar to the actual pump, having a rotor of *unit diameter* revolving at *unit* speed. Since we are free to choose any values we please to serve as standards of speed, head, and diameter, the obvious and simplest plan is to use unity as our standard throughout. Thus :—

Unit speed is a speed of *one revolution per second.*

Unit head is a head of one foot or one metre.

Unit diameter is a diameter of one foot or one metre.

Remembering that all the stipulations controlling changes of variable laid down in earlier paragraphs apply here also, e.g., constancy of speed ratio, flow ratio, width ratio, efficiency, blade angle, etc., we may tabulate performance as follows :—

	Actual Rotor at Working Head and Speed.	Actual Rotor at Unit Speed.	Geometrically-similar Rotor of Unit Diameter Running at Unit Speed.
Speed in revolutions per minute . . .	N	60	60
Speed in revolutions per second . .	$n = \frac{N}{60}$	1	1
Rim velocity . . .	v_2	$v_2 \cdot \frac{60}{N}$	$v_2 \cdot \frac{60}{N} \cdot \frac{1}{d_2}$
Head	H	$H \cdot \left(\frac{60}{N}\right)^2$	$H \left(\frac{60}{N}\right)^2 \left(\frac{1}{d_2}\right)^2 = \frac{H}{n^2 d_2^2}$
Flow velocity .	Y	$Y \cdot \frac{60}{N}$	$Y \cdot \frac{60}{N} \cdot \frac{1}{d_2}$
Area for flow . . .	a_2	a_2	$a_2 \cdot \left(\frac{1}{d_2}\right)^2$
Discharge . . .	$Q = a_2 Y$	$a_2 \cdot Y \cdot \frac{60}{N}$	$a_2 \left(\frac{1}{d_2}\right)^2 \cdot Y \cdot \frac{60}{N} \cdot \frac{1}{d_2} = \frac{Q}{nd_2^3}$

56. Characteristic Performance Numbers. In the above table the expressions that chiefly interest us are those

representing the *head* and the *discharge* of the pump of unit diameter. A slight modification still further heightens their value. By including the term g (acceleration of gravity) in the expression for head, and by using the symbols H, D, as generally representative of head or size, we derive the form :—

$$h_c = \textit{characteristic head number} = \frac{gH}{n^2D^2}.$$

The expression for discharge requires no essential modification, and we may therefore at once write :—

$$q_c = \textit{characteristic discharge number} = \frac{Q}{nD^3}.$$

Both of these expressions are *non-dimensional ;* they are pure numbers, independent alike of the system of units and of the size of the pump. They represent one complete solution of the problem posed in §§ 50 and 51. So long as we are concerned with a given *shape* of rotor, working under its specified conditions of speed ratio, etc., then the characteristic head number and the characteristic discharge number are wholly unaffected by changes of head, speed, and diameter. But changes in the *shape* of the rotor, e.g., changes in the width ratio λ or the blade angle γ, may immediately be reflected in alterations in the characteristic performance numbers. This is graphically shown in Fig. 35, § 60. At last, then, we have two numbers which describe the shape of a pump rotor just as the golfer has a number for describing the shape of his club.

In this book the abbreviations *head number* and *discharge number* will be found convenient. There will never be any confusion between such numbers and the heads and discharges of the pumps themselves ; the one set of values is purely relative, while the other is expressed in units such as feet or metres or cubic feet per second.

57. Specific Speed. The alternative conception of standardised conditions, (ii), § 55, depends upon the fundamental notion of a pump as an apparatus for transferring energy to the liquid flowing through it, § 1. According to this view, a pump is of standardised size if, when generating unit head, it delivers energy to the liquid at unit rate, viz. *at the rate of one horse-power.* The (imaginary) wheel is termed the *specific wheel* and

its speed is termed the *specific speed*. In the following tabular derivation the subscript $_s$ relates to "specific" conditions :-

	Actual Rotor at Working Head and Speed	Actual Rotor at Unit Head	Specific Rotor Running at Specific Speed and giving Unit Head
Speed in revolutions per minute	N	$\frac{N}{\sqrt{H}}$	$N_s = \frac{N}{\sqrt{H}} \cdot \frac{d_2}{d_s}$
Head	H	1	1
Flow velocity	Y	$\frac{Y}{\sqrt{H}}$	$\frac{Y}{\sqrt{H}}$
Area for flow	a_2	a_2	$a_2 \left(\frac{d_s}{d_2}\right)^2$
Discharge	$Q = a_2 Y$	$a_2 \frac{Y}{\sqrt{H}}$	$a_2 \cdot \frac{Y}{\sqrt{H}} \cdot \left(\frac{d_s}{d_2}\right)^2 = \frac{Q}{\sqrt{H}} \cdot \left(\frac{d_s}{d_2}\right)^2$
Power output	$\frac{QwH}{k_p}$	$\frac{\frac{Q}{\sqrt{H}} \cdot w \cdot 1}{k_p}$	$\frac{\frac{Q}{\sqrt{H}}\left(\frac{d_s}{d_2}\right)^2 \cdot w \cdot 1}{k_p}$

By rearrangement of the expressions in the last column of this table, we find that :—

$$\text{Specific speed} = N_s = \frac{N}{\sqrt{H}} \cdot \frac{d_2}{d_s} = \frac{N}{\sqrt{H}} \sqrt{\frac{Q}{\sqrt{H}} \cdot \frac{w}{k_p}}$$

$$= \sqrt{\frac{w}{k_p}} \cdot \frac{N\sqrt{Q}}{H^{5/4}} \quad . \quad . \quad . \quad (5\text{-}1)$$

where :—

N, H, and Q are general terms describing the working performance of the actual pump,

w is the density of the liquid,

and k_p is the horse-power equivalent, or energy per second corresponding to one horse-power (550 ft. lb./sec. or 75 kg. m./sec.).

58. Practical Equivalents for Specific Speed. In most pumping problems the liquid is water—or at any rate it can be

assumed to have the density of water. On this assumption the specific speed formula may be simplified thus :—

$$N_s = K_s \cdot \frac{N\sqrt{Q}}{H^{\frac{3}{4}}} \quad . \quad . \quad . \quad (5\text{-}2)$$

where the factor K_s depends only on the units chosen. This factor can if necessary also take care of the conversion from the mathematically consistent units implicit in formula **(5-1)** to industrial units. Values commonly needed are :—

Units of Speed, N.	Units of Discharge, Q.	Units of Head, H.	Value of K_s.
Revolutions per minute	Imperial gallons per minute	Feet	**0·0174**
Revolutions per minute	U.S. gallons per minute	Feet	**0·0159**
Revolutions per minute	Litres per second	Metres	**0·1155**
Revolutions per minute	Cubic metres per second	Metres	**3·65**

Naturally the specific speed N_s will have a different numerical value if unit head is 1 ft. from what it has if unit head is 1 m.; consequently if there is likely to be any doubt the full designation should be given, e.g. :—

Specific speed = 80 (foot).

or specific speed = 355 (metric).

The rule for conversion is : To convert N_s (foot) to N_s (metric), multiply by 4·44.

Effect of Change of Density. If we know the specific speed N_{sw} of a wheel when working with water, as computed by the foregoing rules, and we want to know the specific speed N_{s0} when the liquid is manifestly not of the same density as water, then evidently from § 57 we can write :—

$$N_{s0} = N_{sw}\sqrt{\frac{w_0}{w_w}}.$$

where the subscript $_w$ relates to water and the subscript $_0$ to the new liquid.

Nominal Specific Speed. In place of the true specific speed as calculated above, engineers are often content to use simply the expression :—

$$N_{sn} = \text{nominal specific speed}$$

$$= \frac{\text{speed in r.p.m.}\sqrt{\text{discharge}}}{(\text{head})^{\frac{3}{4}}} \quad . \quad . \quad (5\text{-}3)$$

Naturally its numerical value will change with every change of the units involved; but it is plain from the above table that the factor K_s will at once provide the connection between the true specific speed and the nominal specific speed, thus :—

$$N_s = K_s \,.\, N_{sn}.$$

Graphical Aids to Computation. The Charts I and II facing page 480 will assist in various sorts of computation and conversion.

59. Characteristic Shape Number. If the true specific speed of a rotor has been correctly calculated, then the resulting numeral will in fact comply with the definition, § 57: "The specific speed of a given rotor is the speed in revolutions per minute of a geometrically-similar rotor of such a size that it will transfer one horse-power to the liquid when generating unit head". Yet the numerical value is very much at the mercy of systems of units, of liquid properties, and so on, § 58; and in truth the mathematical "dimensions" of the expression are highly complex. Fortunately the introduction of the term g brings about just such a simplification as it did when applied to a rotor working under "unit" conditions, § 56. If we write :—

$$n_s = \textit{characteristic shape number} = \frac{1000n\sqrt{Q}}{(gH)^{\frac{3}{4}}}, \quad (5\text{-}4)$$

then the expression fulfils all the requirements of § 51; it is truly non-dimensional, it is independent of the size of the rotor or the properties of the liquid, and it does not depend upon the units *so long as these are consistently chosen*, thus :—

Units of Speed, N	Units of Discharge, Q	Units of Head, H.	Acceleration of Gravity, g
Revolutions per second	Cubic feet per second	Feet	32·2 ft./sec.
Revolutions per second	Litres per second	Decimetres	98·1 dm./sec.
Revolutions per second	Cubic metres per second	Metres	9·81 m./sec.

The reason why the factor 1000 has been introduced is to lift the numerical values of the characteristic shape criterion into a range of whole numbers—otherwise there might be inconvenient fractional values such as 0·07.

The basic formula (**5-4**) can readily be converted for practical use into a simplified form comparable with the expression (**5-2**), § 58, viz. :—

$$\textit{Shape number} \quad n_s = K_{ns} . N \frac{\sqrt{Q}}{H^{\frac{3}{4}}} \quad . \quad . \quad . \quad (\mathbf{5\text{-}5})$$

where the conversion factor K_{ns} has the values :—

Units of Speed, N	Units of Discharge, Q	Units of Head, H	Value of K_{ns}
Revs per min	Imperial gallons per minute	Feet	0·0637
Revs per min.	U S gallons per minute	Feet	0·0583
Revs per min	Litres per sec.	Metres	0·0950

It is illuminating to observe how the *shape number* (a convenient abbreviation) can be expressed in terms of the original descriptive ratios developed in § 52, thus :—

$$\text{Speed } n = \frac{\phi\sqrt{2gH}}{\pi D}.$$

$$\text{Discharge } Q = \pi D\lambda D . \psi\sqrt{2gH}.$$

$$\text{Therefore } \textit{shape number } n_s = \frac{1000\phi\sqrt{2gH}\sqrt{\pi D\lambda D\psi\sqrt{2gH}}}{\pi D(gH)^{\frac{5}{4}}}.$$

$$= 950\ \phi\sqrt{\lambda\psi}.$$

By using the conversion factors given in § 58, the following connection between true specific speed and characteristic shape number may be established, assuming the liquid to have the density of water :-

$$N_s\ (\text{foot}) = 0{\cdot}273\ n_s,$$

$$N_s\ (\text{metric}) = 1{\cdot}213\ n_s. \qquad \textbf{(Example 7)}$$

60. General Significance of Specific Speed and of Shape Number. Although the reasoning that has been carried

through this chapter has been applied only to radial-flow impellers suited for centrifugal pumps, it is equally valid for other types of rotor, e.g., mixed-flow or axial-flow. Indeed the great advantage of the treatment is that it gives the chance of

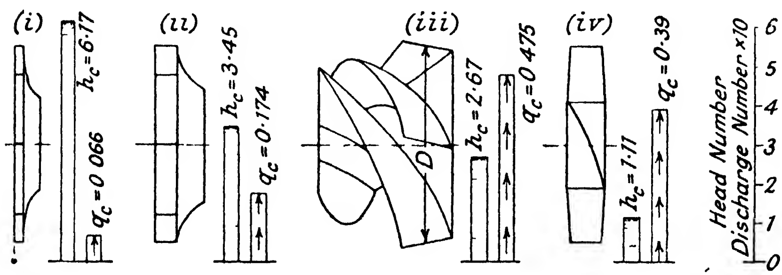


FIG. 35.—Performance of rotors of unit diameter running at unit speed.

quite comprehensive comparisons, such, for example, as those illustrated in Figs. 35 and 36. The diagrams show the continuous change in the behaviour and proportions of rotors as we progress from a "low specific-speed type"—a narrow

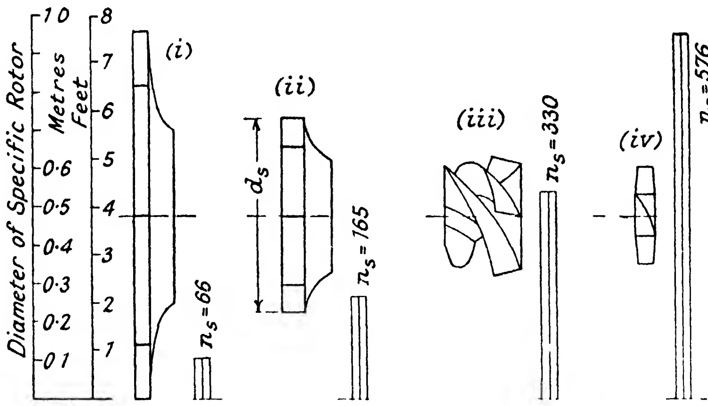


FIG. 36.—Performance of rotors geometrically similar to those shown in Fig. 35, but each transmitting one horse power at unit head.

centrifugal pump rotor—to a "high specific-speed type"—a propeller-type rotor. Together with intermediate types, these cover the entire range of shapes in general use. The shapes selected have the following characteristics :—

Reference Number.	(i)	(ii)	(iii)	(iv)
Description.	Centrifugal.	Centrifugal.	Screw (§ 30).	Propeller (§ 35).
Width ratio λ	0·040	0·107	—	—
Speed ratio ϕ	0·90	1·20	1·35 (mean)	2·10
Flow ratio ψ	0·15	0·20	0·30 (at inlet)	0·41
Head number h_c	6·17	3·45	2·67	1·11
Discharge number q_c	0·066	0·174	0·475	0·39
Shape number n_s	66	165	330	576
True specific speed (metric) N_s	80	200	400	700
Diameter of specific rotor, metres	0·95	0·54	0·29	0·25
Diameter of specific rotor, feet	7·6	4·3	2·3	2·0

61. Connotations of the Shape Number. Fig. 35, then, serves as a graphical statement of the system of standardisation suggested in § 55 (i): performance under *unit conditions*. All the four rotors are of unit diameter, and they are imagined to revolve at unit speed; thus the head numbers and discharge numbers shown to scale by the vertical columns represent the actual behaviour of the pumps. The type (i) impeller generates by far the highest head, and the axial-flow pump (iv), by far the lowest head; but the type (iv) rotor handles a relatively much larger quantity of liquid. It is illuminating to note how this agrees with the numerical comparison worked out in § 40.

The rotors depicted in Fig. 36 are respectively identical in shape with those of Fig. 35, but they are of such a size that each will transmit energy to the liquid at the rate of one horse-power, when they generate unit head—that is, they are "*specific*" *rotors*, § 57. The dimension scales, and the figures in the table above, show that for a unit head of one foot a given type of wheel is much bigger than if unit head is one metre. The heights of the columns are proportional both to the shape number and to the specific speed of the respective wheels; they make it clear that although in proportion to its size the axial-flow rotor seems to be the most effective energy transmitter, yet it has to run very fast.

It has been mentioned that the term H in expressions for pump performance is generally representative of the head generated. For demonstrating the mathematical foundation of the expressions, the ideal head created by the rotor has hitherto been taken as H, but for describing the working conditions of actual pumps the effective head H_e should be used, § 163. Nevertheless it is still useful to think in terms of, e.g. "specific speed per rotor", because when dealing with pumps having multiple rotors, § 115, one must be particularly careful to specify the behaviour of *individual* rotors, and *not* of the complete pump.

PART B

DESIGN AND CONSTRUCTION

CHAPTER VI

GENERAL PROBLEMS OF PUMP DESIGN

62. Practical Procedure. Part A of this book has presented some of the basic principles which must inevitably guide the pump designer. Now in Part B these principles are offered in another form—an empirical form that has been developed by the testing and continuous running of actual pumps.

In general, the chapters of Part B will continue to apply to the pump only rather than to the complete installation; their scope will not extend beyond the pulley or half-coupling on the shaft. It is Part D of the book which will concern itself with the final problem of assembling into an appropriate pumping set the various elements involved, e.g., the pump itself, the engine or motor that drives it, and the piping and valves through which the liquid flows. This present chapter deals with general matters common to all pumps: then there follow chapters devoted to specific types of pumps, e.g., centrifugal pumps for clean cold water; screw pumps, etc.; multiple-rotor pumps; and finally machines specially built to meet special requirements. A description of the constructional features of each type will prepare the way for a set of design formulæ adapted for immediate practical use.

63. Prerequisite Data. Certain essential items of information must be available before a suitable pump can be chosen or designed. They may be summarised thus :—

Liquid. The liquid to be pumped will be . . . (water, oil, spirit, acid, etc., etc.).

Its temperature will be . . . (° F. or ° C.).

Its density will be . . . w (lb./cu. ft., kg./lit., etc.), or its specific gravity will be. . . .

Its viscosity will be . . . (poises, secs. Redwood, etc.).

It will be clean (or dirty, or gritty, etc., etc.).

Rate of Discharge. The quantity to be delivered in unit time will be . . . (gall./min., lit./sec., etc.).

Head or Pressure. The liquid is to be lifted through a height of . . . (feet, metres); or is to be forced against a pressure of . . . (lb./sq. in., etc.); or is to be pumped through . . . (feet of piping, etc.).

With the help of these figures, it is required to find out :—

(i) Whether a centrifugal, screw, propeller, etc., type of pump is indicated.
(ii) What its speed should be.
(iii) What power will be needed to drive it.
(iv) How the machine is to be selected, designed or built.

A preliminary step is to put the information in the right form. The rate of discharge must be converted into appropriate units, e.g. cubic feet per second, pounds per second, or the like, § 51; and by the methods explained in § 163 the *effective head* H_e must be estimated, at least in a tentative manner. If for any reason the diameter of the pump branches is involved, then the table in § 87 may be helpful.

64. Pump Type, Speed, and Power. If the liquid is clean cold water, and if there are no subsidiary stipulations such as those detailed in the next paragraph, then the value of the *effective head* H_e will point fairly conclusively to the proper type of pump. Experience shows that the main categories of pump should preferably be restricted to the following ranges of head :—

Head		Type of Pump.
Feet	Metres	
Above 20	Above 6	Centrifugal (Chap. VII)
10 to 50	3 to 15	Screw or half-axial (Chap. VIII)
5 to 20	$1\frac{1}{2}$ to 6	Axial-flow (Chap. VIII)

The overlap in these ranges shows that the figures alone are not decisive; the final choice of type may be governed by various secondary factors, § 65.

For each type of pump there is an appropriate range of *shape number* or *specific speed* (Chap. V), thus :—

Type of Pump.	Shape Number.	True Specific Speed.	
		Foot.	Metric.
Centrifugal . . .	60-330	16- 90	70- 400
Screw or half-axial . .	250-500	70-140	300- 600
Propeller (axial-flow) .	400-800	110-220	500-1000

Within the range of shape number for each type, it is probable that the *higher* the head, the *lower* will be the most suitable shape number to choose. An example of this trend or tendency (for it is hardly more than that) is illustrated in Fig. 52, § 90.

Having, by the use of some such graph or in any other way, fixed upon some likely value to suit the given conditions, it remains to determine the *rotational speed* N of the pump shaft in r.p.m. We know the effective head H_e, the discharge Q, and the shape number $n_s = \dfrac{1000 \cdot \dfrac{N}{60}\sqrt{Q}}{(gH_e)^{\frac{3}{4}}}$ (§59) ; so the desired value of N can at once be extracted. Useful computation charts face p. 480.

Empirical information is again available (§ 91) for estimating the *gross* or *overall* efficiency of the pump, η_m, and this in turn permits the necessary *shaft-horse-power input* P_s to be assessed, thus : $P_s = \dfrac{QwH_e}{K_p \cdot \eta_m}$. . . . . (§ 165)

(Examples 8, 9)

65. Supplementary Data. It is hardly likely that the smooth routine of selection just outlined can invariably be followed. Here are some of the factors that may throw it out of gear :—

(i) *Shaft Speed fixed by Motive Unit.* If the pump is to be direct-coupled to an engine, electric-motor, or steam turbine whose speed is already determined, then the shape number of the pump rotor is similarly fixed. This information may settle, without further argument, what class of pump is alone admissible, § 64. If the computed shape number or specific speed

falls outside customary limits, then the pump may have to be fitted with multiple rotors, Chapter IX.

(ii) *Conditions of Service.* The location of the pump—whether at ground level or underground; or the kind of liquid—whether cold water or hot oil or acid—may influence the choice of pump, Chap. X. So also may the conditions of performance, as detailed in Part C of the book, Chaps. XIII, XVI.

(iii) *Commercial Restrictions.* Small rotodynamic pumps are built for sale in a highly competitive market, from which it follows that purely hydraulic considerations may have to give way to the necessities of economical production. As it would be quite impracticable to design and construct a special pump to suit each individual customer's desires, a range of standard types must be evolved from which the client can be offered the one that most nearly suits him. Inexpensive modifications to standard components—rotors, casings, etc.—will quickly result in a satisfactory machine, even if one cannot be delivered from stock. But evidently the original designing must be skilfully done so as to ensure the greatest degree of adaptability, e.g. see §§ 70, 71, 301-303.

66. Constructional Elements. Rotodynamic pumps are built up from the following primary components:—

Revolving

(i) The rotor or bladed wheel (impeller, propeller, etc.), Chaps. I, III.
(ii) The shaft on which the rotor is mounted. Sometimes there may be two or more rotors on the same shaft.

Stationary

(iii) Inlet and outlet passages for leading the liquid into and out of the pump.
(iv) The recuperating device which receives the liquid discharged from the rotor (diffuser, volute, etc.), Chapter IV.
(v) Bearings for the shaft.
(vi) The casing, bedplate, or frame which supports the various fixed elements.

Sometimes two or more of these functional elements are combined in a single casting. Thus in small pumps two castings only will serve for all the items (iii) to (vi).

Materials of Construction. (*) For normal services, cast iron can be used both for the casing and for the rotor ; except for the steel shaft, that is, virtually the entire pump is of cast iron. As the liquid grows progressively more corrosive, or as greater durability is required from the pump, we find sleeves or liners of non-ferrous metal being used, or the rotor and shaft being made entirely of bronze. For extreme conditions the complete pump casing may be of cast or forged steel, or of gun-metal or bronze ; or the internal parts may be of monel metal, etc., etc. Alternatively there may be an internal lining of lead, hard rubber, etc., Chapter X.

67. The Rotating Components. In designing the *rotor* the first step is to fix the diameter and width so that the *velocity of flow* is kept within suitable limits ; the wheel, that is to say, must be made big enough to pass the specified amount of liquid. Usually there is a fairly well-defined connection between the shape number of the rotor, and the flow ratio, § 60 ; moreover, a given shape number often corresponds to certain relations between width, diameter, etc. From these empirical relationships, therefore, a provisional impression of the shape and size of the wheel can quickly be derived.

The next step is to find out the shape of blades and the number of blades which will enable the wheel to transmit to the liquid the stipulated quantity of energy.

Although the primary duty of the *shaft* is to transmit to the rotor the power received from the motive unit (engine, motor, etc.)—from which it follows that torsional stresses will inevitably be set up—yet the shaft has to withstand other stresses as well. The rotor may be subjected to (i) transverse unbalanced hydraulic loads which tend to bend the shaft, and to (ii) axial hydraulic loads which tend to displace the whole rotating element longitudinally. If the rotor is out of balance, too, there will be a further transverse dynamic load due to the tendency of the shaft to " whirl ". Finally there may be externally-applied loads such as result from belts, gears, or misalignment of couplings.

68. The Stationary Components. The passages of the *volute, diffuser*, etc., must not only be designed to carry out the necessary energy transformations as economically as possible, but they must be mechanically robust enough to withstand the

internal pressure of the liquid. Additional stresses may be transmitted to the casing from the suction and delivery pipes that are bolted to it ; thermal expansion and contraction of the pipes, or unskilful installation, will tend to deform the pump casing. After the pump has been taken into service it will be a great advantage if the casing can be opened up for inspection of the rotor, etc., without breaking too many joints and without disturbing the pipe connections.

The *bearings* must be proportioned for continuous running when the shaft is subjected to the worst combination of transverse and axial loads. It may be necessary to allow for deformation of the casing which supports the bearing housings.

Leakage must be carefully controlled. Where the shaft passes through the walls of the casing, glands and stuffing-boxes are required to keep liquid from leaking *out* of the pump, or to prevent *air* from leaking *in*. Internal leakage of liquid from the high-pressure to the low-pressure parts of the pump must be brought within permissible limits.

CHAPTER VII

CONSTRUCTION OF STANDARD CENTRIFUGAL PUMPS

69. Types of Impeller. These may be classified into :—

(i) Single-entry (side-entry, single-suction) impellers.

(ii) Double-entry (double-suction, balanced suction) impellers.

Another mode of subdivision uses the titles :—

(*a*) Closed (shrouded) impellers, and

(*b*) Open (skeleton) impellers.

Examples are illustrated in Fig. 37.

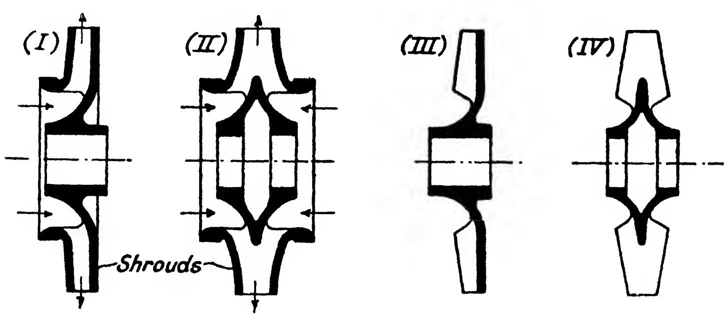


Fig. 37. Cross sections of typical impellers.

(I) Here is a closed, single-entry impeller. Circular *shrouds* or discs or side plates guide the liquid, support the

blades and transmit to them the torque which enables them to give tangential acceleration to the liquid. The liquid enters from one side only.

(II) The liquid can enter this closed double-inlet impeller from both sides. A higher specific speed can thus be realised, and the problem of balancing end thrust is simplified.

(III) This side-inlet open impeller has a shroud on one side only. The unsupported edges of the blades work with a small clearance against the cheek of the fixed casing.

(IV) The blades of this double-entry skeleton impeller are supported only by the boss or hub of the casting.

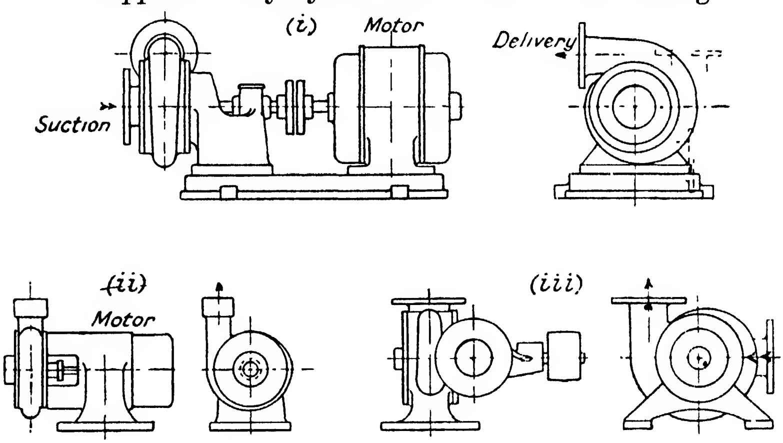


FIG. 38.—Types of side-inlet centrifugal pumps.

Closed or shrouded impellers are nearly always preferred for normal pumps; the advantages of the open types are more apparent when impure liquids are in question. When, as in the present instance, the pump is intended for cold and moderately clean water, the impeller is almost invariably formed of a one-piece casting in iron, bronze or gun-metal.

(For comments on conventional systems of projection for impellers, see § 99.)

70. General Disposition of Pump. (*) Representative types of pump to suit both side-inlet and double-inlet impellers, with either horizontal shaft or vertical shaft, may be compared in Figs. 38, 39, and 40. The differences in design will serve to illustrate the points raised in § 65 (iii) concerning manufacturing

convenience and possibilities of modifying stock arrangements to suit customers' requirements. All the models have *volute* casings.

Side-inlet Horizontal Pumps. In type (i), Fig. 38, the delivery flange is cast integrally with the volute, and the casting is spigoted to a frame which also carries an outer bearing. The suction pipe can be bolted to a flange mounted axially on the volute cover. By turning the volute on its spigot through successive angles of 90°, the delivery flange can be made to point upwards or downwards, to the left or to the right, as the client may direct. The little flange-mounted pump (ii), is similarly adaptable ; screwed connections suffice for the pipes of 1-in. diameter or so. A specially-built electric motor carries the volute casting. These " close-coupled " electrically-driven units may also be obtained in larger sizes.

In type (iii), Fig. 38, the feet of the pump are cast integrally with the volute, and thus for a given casting no variation in the delivery flange position is permissible ; but the suction branch can be swivelled round as desired, the outer bearing housing being arranged accordingly. Details are shown in Fig. 44 (ii).

71. Double-inlet Horizontal Pumps. The overhung type (i), Fig. 39, rather resembles arrangement (i), Fig. 38. Here also the volute with its branches can be swivelled on its spigot, but now the branches lie in the same plane and there is an unvarying angle—in this case of 180°—between them. In the *split-casing type* (ii), no departure from the standard flange arrangement is possible, but on the other hand the whole casing is exceptionally stiff and solid. Moreover, the shaft is well supported by two bearings which work in excellent conditions, quite away from the liquid. A final reason for the great popularity of the split-casing type is the accessibility of the impeller. It can be inspected immediately the top half of the casing is taken off ; and if the top halves of the bearing shells are also removed, the impeller and shaft can readily be lifted out. It is true that no pipe joints need be broken to inspect the impellers of the pumps shown in Fig. 38 (iii) and Fig. 39 (i), but the removal of the rotating elements is a good deal more troublesome than it is in the split-casing pump.

The volute of the low-lift pump, Fig. 39 (iii), is in two halves with a transverse joint ; the two parts are identical except for

the "handing". Although the impeller cannot be taken out without virtually dismantling the entire pump and pipework, yet the simplicity and solidity of the design fits the pump very well for irrigation and similar rough duties.

In point of size we have now progressed from the little unit represented in Fig. 38 (ii), up to medium-sized pumps having branches of 12-in. diameter or more and capable of discharging several thousands of gallons per minute each. In the double-inlet pumps (i), (ii), and (iii), the separation of the incoming liquid into two equal streams, one for each side of the impeller, has been carried out within the casing itself, Figs. 39 and 44 (i).

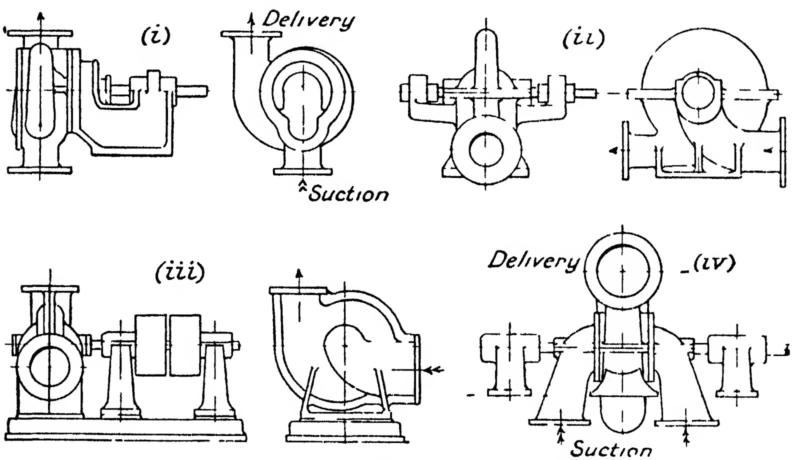


FIG. 39.—Types of double-inlet horizontal pumps.

For still larger units this may no longer be desirable. Since in any event the casing is too big to be formed of a single casting, but must be bolted together from separate sections, the opportunity may be seized to provide individual suction branches for each side of the impeller, Fig. 39 (iv). Some of the very largest pumps in use to-day follow these general lines.

72. Vertical-spindle Pumps. Still keeping within the range of what can be regarded as standard commercial products, we come to dispositions which often have the particular virtue of compactness. (*) This is especially apparent in the motor-driven set, Fig. 40 (i), which takes up a floor space only equivalent to the area of the circular base. Although the casing is split axially, viz. in a vertical plane, yet the impeller is of the

side-inlet type, the liquid entering from above through an inlet passage that resembles the one shown in Fig. 38 (iii). If, in this latter model, the inlet bend were swung round so as to point vertically upwards, then the similarity between the two designs would be manifest. A larger side-inlet pump of high specific speed is illustrated in Fig. 40 (ii), but here the water enters from below.

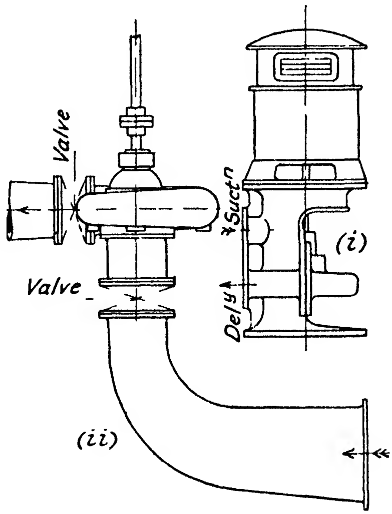


FIG. 40.—Vertical shaft pumps

Relatively slight modifications to double-inlet split-casing pumps, Fig. 39 (ii), permit these to be laid on their side and used as vertical-spindle units.

73. Forces Acting on the Rotating Elements.

Having now a general impression of how the pump parts are disposed, we can return to a more detailed study of the forces acting on the shaft and impeller in order to design the shaft and its bearings, § 67. These forces include :—

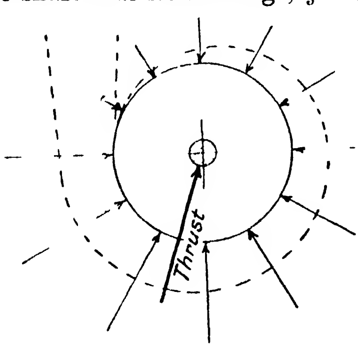


FIG. 41 — Uneven pressure distribution around volute.

(i) *Weight.* Clearly the entire weight of the revolving elements, comprising shaft, impeller, and half-coupling or pulley, etc., must be supported by the bearings.

(ii) *Transverse Loads.* (*a*) *Dynamic.* Should the impeller casting be out of balance, due to initial inaccuracy of construction, or should corrosion or uneven wear during service create this condition, then when the pump is running the effect of centrifugal force on the revolving mass will be to apply a transverse load on the shaft. The effect will be intensified if the eccentricity increases as the shaft deflects.

(*b*) *Mechanical.* Driving belts, or imperfectly-aligned couplings or distortion of the frame or casing may set up side loading.

(*c*) *Hydraulic.* Uneven pressure distribution around the periphery of the impeller may generate a resultant transverse thrust. If, for example, the pump has a variable-velocity volute, § 45 (*a*), then necessarily the pressure will rise continuously from the tongue of the casing all the way round to the outlet ; for it is the whole purpose of the recuperator to generate such a pressure increment. The effect of these increments is suggested in Fig. 41, which clearly shows how the resultant thrust must fall on the shaft Even though the volute design is such that the thrust is not developed during normal pump

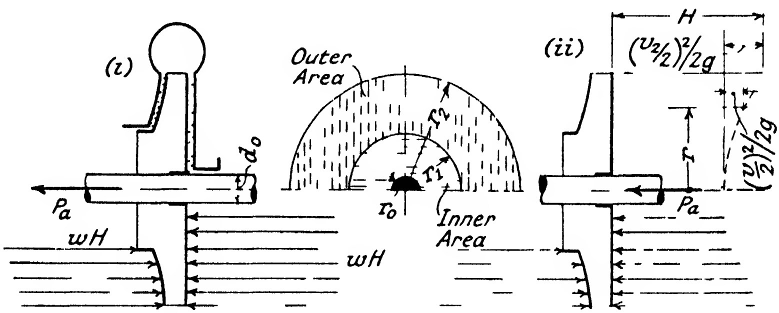


FIG. 42 —Static pressure distribution on stationary and moving impellers

operation, it may and almost certainly will arise during reduced flow and increased-flow conditions, § 207 (i).

(iii) *Axial thrusts*, as detailed in the following paragraphs

74. Axial Hydraulic Thrust on the Impeller. (i) *Static.* Let us first examine the hydraulic thrust on a dummy side-inlet impeller at rest, Fig. 42 (i). We imagine that a circumferential band prevents return flow through the impeller passages, and that the pressure is maintained at its normal value by means of some other pump discharging into the volute. The liquid, like the wheel, is at rest, and the uniform pressure-head H prevails throughout the zones indicated by stippling. As we are here concerned with the static pressures acting on the outer faces of the impeller shrouds or discs, it will be convenient to distinguish between the *outer* area of the discs and the *inner* area. Their limits are clearly distinguished by

hatching in the diagram. Evidently the equal and opposite thrusts on the two outer areas completely neutralise one another, but on the inner area there is a resultant thrust, acting *towards* the impeller "eye" or entry, having the value

$$wH \cdot \frac{\pi}{4}(d_1^2 - d_0^2),$$

when d_0 represents the shaft diameter, and d_1 the impeller inner diameter.

Now, with the volute pressure maintained unchanged, we imagine the impeller to be rotated at its working speed. What will happen to the liquid contained in the clearance spaces between shrouds and casing ? It seems a fair assumption that the liquid elements in contact with the shrouds will be carried round at the same speed as the impeller, while the elements touching the stationary casing will remain at rest. The *average* speed of the elements at a given radius, therefore, will be one-half of the tangential velocity of the impeller at that radius. The effect will be the same as if forced vortex motion were set up, with the result that pressures in the clearance spaces are everywhere *less* than the original pressures when the impeller and liquid were at rest. At any radius r, the reduction will have the value

$$\frac{\left(\frac{v_2}{2}\right)^2}{2g}\left(1 - \left(\frac{r}{r_2}\right)^2\right).$$

where v_2 represents the impeller rim velocity.

Over the *outer* areas of the impeller shrouds, equilibrium will still hold good ; over the inner area there will still be a resultant thrust, but of less intensity than before. The resulting pressure-distribution will thus be of the type suggested in Fig. 42 (ii). By an integration process, it is found that the resultant unbalanced thrust has the value :—

$$P_a = w \cdot \pi(r_1^2 - r_0^2)\left[H - \frac{\left(\frac{v_2}{2}\right)^2}{2g} \cdot \left(1 - \frac{r_1^2 + r_0^2}{2r_2^2}\right)\right] \quad (7\text{-}1)$$

The conditions in the actual pump, when the real impeller is itself generating the pressure in the volute, can hardly be very

different from those just assumed; consequently we may accept equation (7-1), at least for provisional calculations, inserting for safety the effective head on the pump H_e as equivalent to the head in the volute (§ 163). **(Example 10)**

There remain two other items which may effect the static hydraulic axial thrust on the shaft. Experiments and common reasoning show that the mean angular velocity which the impeller imparts to the liquid in the clearance spaces is influenced by the roughness of the surfaces of shrouds and of casing, and also by the axial clearance distance between these surfaces. Any lack of symmetry in these respects will affect the pressure distribution and may thus be responsible for a differential hydraulic thrust contributed by the *outer* areas, Fig. 42. This would happen if the impeller were incorrectly centred in an axial direction between the two faces of the casing. Finally there might arise a differential thrust on the shaft itself. Suppose, for example, that in the pumps shown in Figs. 39 (ii) and 39 (iv), the shaft at the point of emerging through the two stuffing-boxes was of different diameters; then the resulting axial load would be represented by the difference in shaft cross-section multiplied by the *suction* pressure, positive or negative as the case may be.

75. Hydraulic Axial Thrust. (ii) *Dynamic.* Further axial loads may fall upon the impeller by reason of the deflection imposed on the liquid elements flowing through it. Force is needed to divert these elements from their original paths. We should be very conscious of that, remembering that deflection in a tangential direction is the basic principle of all rotodynamic pumps. In the conditions studied in Chapters I and II, the deflecting force was applied by "cylindrical" blades whose surfaces were everywhere tangential to planes parallel to the pump axis; there could be no *axial* component of the force.

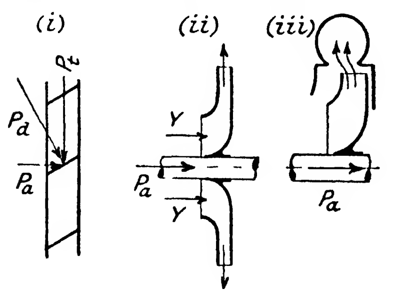


FIG. 43.—Examples of dynamic axial-thrust.

(*a*) But now consider the type of blade sketched in Fig. 43 (i). Evidently the reaction of the liquid on the blade, P_d, § 26, will certainly have an axial component P_a, and the sum of all these components on all the blades will mount up to a quite substantial thrust. This dynamic axial thrust will be still more pronounced in the mixed-flow, high specific speed impeller described in § 99.

(*b*) In an ideal side-inlet impeller, Fig. 43 (ii), the whole incoming stream of liquid is deflected through a right-angle. Thus if Y is the original axial velocity, and W the weight of liquid per second, evidently the axial dynamic thrust will have the value $P_a = \left(\frac{W}{g}\right)Y$.

(*c*) Axial displacement of the impeller relative to the volute, i.e. lack of symmetry in locating the impeller endwise, may set up dynamic axial loads, just as it promoted static hydraulic loads, § 74. We can be sure that if the liquid stream leaving the impeller is forced over sideways, Fig. 43 (iii), there will result a local inequality of pressure over the *inner* faces of the shrouds. The corresponding differential thrust P_a will ultimately have to be carried by the thrust bearing.

76. Methods of Relieving Axial Thrust. If it be contended that the division of the various items of hydraulic thrust into static and dynamic loads is rather arbitrary, we can soon find another classification which may be more helpful. The load associated with what we have called static pressure evidently depends upon the *head* generated by the pump; the load due to dynamic pressure, § 75, depends upon the *discharge* of the pump; while the loads due to constructional irregularities are by their nature quite unpredictable. The net result is that the gross axial thrust transmitted to the bearings may vary widely in the course of the diversified duties of the pump. What equal and opposite force can we generate that will exactly keep pace with these variations? There are various solutions:—

(i) *Twin Impellers or their Equivalent.* Why not mount on the same shaft as the original side inlet impeller another identical one, but *facing in the opposite direction?* If the rate of flow of liquid through the two wheels is also identical, then the net axial thrust on the shaft will be zero. In pumps

with multiple impellers we do in fact find this expedient extremely valuable. Sometimes the twin impellers are set eye to eye, with the liquid streams flowing in parallel, § 145; more often the wheels are arranged back to back, and the liquid flows through them in succession, § 119. For the single-stage pumps now under review, the two opposed impellers are merged into a single casting—a casting that is already familiar to us under the name of *double-inlet impeller*, § 69. The manner in which the incoming liquid is divided into the two equal streams that feed the two impeller eyes is suggested in Fig. 44 (i); this

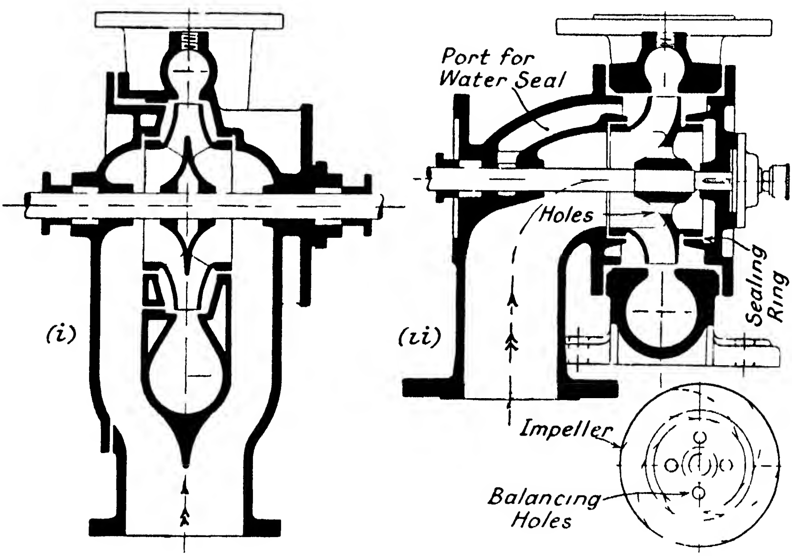


FIG. 44.—Systems for minimising axial hydraulic thrust on impeller

diagram is a sectional view of a typical pump casing such as (i), Fig. 39.

But here again perfection cannot be expected. Not only may inequalities in the impeller casting or the casing passages disturb the symmetry of flow; but even before the main stream is parted, an iregular velocity distribution may have been imposed upon it by a bend, elbow, or similar deviation in the suction pipe adjacent to the pump suction flange. So a small unbalanced thrust on the impeller must always be allowed for. It may even be a large thrust if one of the impeller eyes becomes obstructed for any reason.

77. Relieving the Axial Thrust (*continued*). (ii) *Balancing Holes.* The reason why the attractive balancing principle just described is not universally adopted is that the side-inlet impeller has valuable advantages that we do not want to abandon. Whenever operating conditions are especially severe, due to high temperature or high pressure or aggressive liquids or plurality of impellers, then the designer will be disposed to favour the very simple end-suction principle exemplified in Fig. 38 (i). For the relatively unexacting duties now in question, too, the end-suction arrangement is inexpensive for small pumps and hydraulically advantageous for large pumps such as (ii), Fig. 40.

If, then, we are unable to neutralise under all conditions the axial thrust on the single-inlet impeller, we must try to reduce the thrust. The simplest way of doing this is to drill some holes through the shroud and thus let the pressures on the two sides equalise themselves, Fig. 44 (ii). By itself, this operation would merely provide a loop-way by which large volumes of water would slip back from the delivery to the suction side of the pump ; so to prevent or at least very much reduce this leakage, a sealing ring, § 83, must be cast on the back of the impeller, which can work with a small clearance within a similar ring cast on the casing, Fig. 44 (ii). This device is largely used both for small and for large pumps. (Note that Figs. 38 (iii) and 44 (ii) refer to the same machine.)

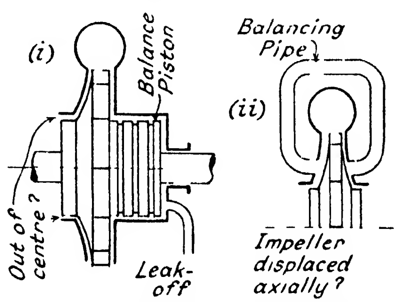


FIG. 45.—Other devices for relieving axial thrust.

(iii) *Balancing Piston.* Very occasionally the axial thrust is relieved by a rotating drum or piston as shown diagrammatically in Fig. 45 (i). A leak-off passage conducts leakage liquid to waste or back to the suction pipe. In this diagram is suggested also one of the effects of the transverse thrust mentioned in § 73 (ii) c ; the shaft is forced out of centre.

(iv) *Balancing Pipe or Port.* Any lack of symmetry in the thrust on the *outer* areas of the shrouds, § 74, whether of

single-inlet or of double-inlet impellers, may be mitigated by a balancing pipe of the sort shown in Fig. 45 (ii), or by an integrally cast port which serves the same purpose.

78. Types of Bearing. Before choosing or designing the bearings which will resist the various loads on the shaft, it is necessary to ask whether the hydraulic balancing devices just described do in fact offer the best solution of the problem. Although effective, these devices entail continuous waste of energy as the liquid leaks away, §§ 83, 192. If, therefore, bearings can be found which can deal with the *unrelieved* gross load transmitted from the impeller, they would provide the most economical answer. In fact, this only happens in small pumps; designers usually prefer to embody hydraulic relieving systems and to load the bearings as lightly as they can.

Types of bearings available are :—

Journal :—

(i) Sleeve with oil ring or oil pump lubrication.
(ii) Sleeve with grease lubrication.
(iii) Ball or roller.

Thrust :—

(iv) Collar with oil or grease lubrication.
(v) Ball.
(vi) Tilting-pad (Michell or Kingsbury).

Combined :—

(vii) Ball or roller.

Cooling devices for heavily-loaded bearings in large pumps include (*a*) water-cooled bearing shells, embodying a jacket through which coolant is circulated, and (*b*) oil-coolers for the oil which is pumped through the bearings

Another distinction is between *external* bearings and *internal* bearings. As the liquid cannot reach external bearings, these work under the same favourable conditions as line-shaft bearings and the like; but internal bearings may be exposed to the liquid passing through the pump.

79. Disposition of Bearings. A renewed examination of the pumps already described, §§ 70, 72, will give an impression of the varied possibilities of supporting the shaft. The numbers relate to the respective outline drawings :

Fig. 38 (i). The side-inlet impeller is overhung. The shaft has one external bearing and one internal bearing; the latter

may either be grease-lubricated or it may be lined with special bronze suitable for water lubrication.

Fig. 38 (iii). This shaft also has two bearings, but the internal grease-lubricated bearing is located in the volute cover, on the outboard side of the impeller, Fig. 44 (ii). Because the impeller is nominally balanced, end thrust will be so light that it can be taken by a plain collar incorporated in the external oil-ring type of sleeve bearing.

Fig. 39 (i). Except for the difference in impeller and inlet and outlet branches, this arrangement is similar to (iii), Fig. 38. The bare shaft end serves as a reminder that in commercial practice a stock machine must be suited either for belt drive or for direct coupling, and the external bearing must be proportioned accordingly.

Fig. 39 (ii). The shaft is well supported here. If the two external bearings are of the ball or roller type, then provision can easily be made for taking residual end thrust. If ring-oiled sleeve bearings are preferred—which often happens—a plain collar will look after the thrust component.

Fig. 39 (iii). As this medium-sized pump receives its power through fast and loose pulleys, a pair of substantial external pedestal journal bearings is disposed so as to take the heavy pull of the flat driving-belt. The two internal bearings on either side of the impeller are more in the nature of guide bearings.

Fig. 39 (iv). External pedestal bearings offer the best solution here also, and for this large pump they must be really well-built.

Fig. 40. In these vertical-shaft pumps the full weight of the rotating member must inevitably come on the thrust bearing, either augmented or diminished by unbalanced hydraulic thrust. A ball-thrust bearing would probably serve for the smaller pump, while a water-cooled Michell tilting-pad bearing would certainly be best for the large pump. Between the overhung impeller of this pump, (ii), and the thrust bearing, there is a white-metal lined steady-bearing.

80. The Shaft. When once the intensities and points of application of the various transverse and axial loads have been estimated, and also the induced reactions from the bearings, then in principle the shaft diameter at various points might be calculated. But in fact it is advisable to add to the dimensions

so computed a generous margin, in order to discourage vibration under the worst conditions. Vibration is not only objectionable in a general mechanical sense, but it intensifies the difficulty of maintaining the gland packings, § 86. So it is experience which must principally guide the designer in securing the necessary stiffness both in shaft and in frame.

Two constructional problems are, (i) how to secure the rotor to the shaft, and (ii) how to protect the shaft from corrosion and from wear. A satisfactory combined solution is illustrated in Fig. 58; the driving torque is transmitted to the impeller through one or more sunk keys, and the impeller is located axially by a sleeve which forces it hard up against a shoulder on the shaft. The assembly is locked by external nuts. The renewable non-ferrous sleeve, and its companion on the other side of the impeller, at the same time guard the steel shaft against rusting and they take the wear which is inevitable at the points where the shaft passes through the stuffing-boxes, § 81. A combined sleeve and locking-nut is suggested in Fig. 49. When conditions point towards a non-ferrous impeller, then the whole shaft may be of bronze.

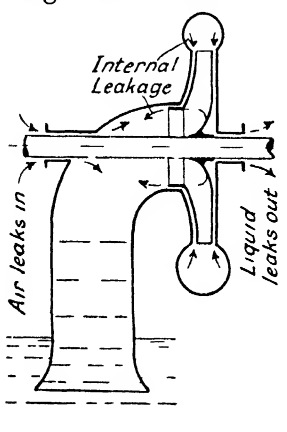


FIG. 46 Internal and external leakage

Overhung impellers may be fixed to the tapered end of the shaft by means of a key and a nut, Fig. 60.

81. Leakage and its Prevention. (*) There are three possibilities of leakage at adjacent surfaces of fixed and rotating parts of the pump, § 68. (i) Liquid may leak internally from the high-pressure to the low-pressure side of the casing, (ii) liquid may leak *from* the high-pressure part of the pump into the external atmosphere, and (iii) atmospheric air may leak *into* those parts of the pump where sub-atmospheric pressure (vacuum, suction) prevails, Fig. 46. Why should any special difficulty be expected here? If ordinary packed glands have given good service in other types of machinery, why should they not do so now? In regard to internal leakage the answer

is soon forthcoming. It is true that a packed gland of abnormal diameter might be provided for the impeller eye, where it fits circumferentially against the pump casing ; but the rubbing speed would be so high that heating and rapid wear might be expected. Besides, the moment of the frictional force would be so relatively great that the gland would consume an excessive share of the power fed into the shaft.

The gland and stuffing-box that prevent the escape of liquid *from* the pump need not be troublesome unless the pressure is comparatively high. But, if, in such conditions, the gland is tightened up more and more in the hope of reducing leakage, then heating and excessive wear may result. Moreover, the packing squeezed hard against the shaft or its protecting sleeve may in time cut a groove ; thereafter, still further screwing-up of the gland nuts will force the packing against the sides of the groove as against a shoulder on the shaft, so creating a sensible addition to the end thrust on the shaft.

But dangers from excessive pressure surely ought not to arise on the suction side of the pump, where the maximum pressure-difference on the stuffing-box cannot possibly exceed 10 or 12 lb./sq. inch ? Here it is not the excessive fluid pressure that might cause trouble, but the pressure between packing and shaft caused by screwing up the gland nuts too tight. If they were left slack, small amounts of air would leak into the pump and might create very tiresome operating troubles, § 358.

These difficulties can be solved with varying degrees of success by methods described in the following paragraphs.

82. Leakage between Impeller and Casing. If, as pointed out above, a packed gland around the impeller eye is ruled out because of wear and friction, and if metallic contact between impeller and casing is inadmissible for the same reason, then we must give up all hope of making a liquid-tight seal at this point. Since we cannot actually prevent leakage, then, we must do the best we can to keep down the rate of leakage as low as possible. But already the mere acknowledged existence of leakage brings a new complication into the design of the pump. The impeller must be capable of passing not only the stipulated quantity of liquid Q that issues from the delivery pipe, but it must also handle the additional quantity q_l that

leaks back through the annular clearance space. This added quantity q_l, which may amount to anything from 3 to 10 per cènt of Q, is termed the "leakage loss" or "slip loss" or "short-circuit loss".

The hydraulic principles that will assist in reducing the leakage loss are represented in Fig. 47 (i). As the liquid flows from a high level to a low level through a circular pipe with mean velocity v_e, its energy H is dissipated in three ways, viz.: (i) an eddy loss h_l at the sharp-edged entry, (ii) friction loss h_f in the pipe itself, and (iii) an eddy loss h_v at outlet. By substituting in the basic equation $H = h_l + h_f + h_v$ the customary values, we arrive at the expression: —

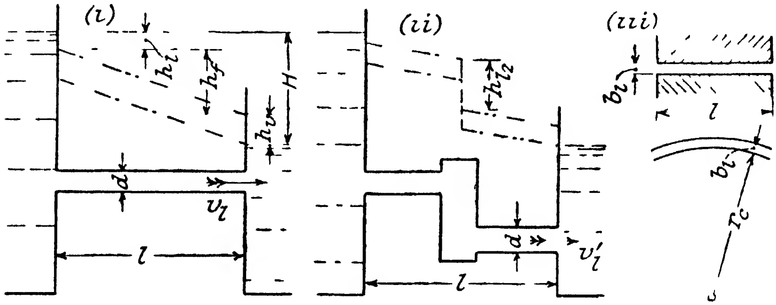


FIG. 47.—Flow through leakage passages

$$H = C_l \cdot \frac{v_l^2}{2g} + \frac{4fl}{d} \cdot \frac{v_l^2}{2g} + \frac{v_l^2}{2g}.$$

To ensure the *minimum* rate of flow through the pipe, we ought evidently to make the pipe as small as possible or as long as possible. But even if the energy loss H, the length l, and the diameter d were fixed, the flow could still be reduced by generating increased turbulence, e.g., by abrupt changes of direction as suggested in Fig. 47 (ii). In this way an amount of energy h_{l2} is dissipated.

83. Theory of the Sealing Ring. Comparing now the passages (i) and (ii) with the actual pump, Fig. 47 (iii), we observe that a circular pipe is replaced by an annular space of mean radius r_c, length l and radial width b_l. Diagram (iii), in fact, represents an enlarged view of the running clearance between casing and impeller, Fig. 46. The pressure head difference between the two ends of the clearance space can be

estimated by the method of § 74, that is to say, it will have the value

$$H_l = H_e - \frac{\left(\frac{v_2}{2}\right)^2}{2g}\left[1 - \left(\frac{r_1}{r_2}\right)^2\right]$$

The value corresponding to pipe diameter d, to be inserted in the friction term h_f, is now $2b_l$, hence $h_f = \frac{4fl}{2b_l} \cdot \frac{v_l^2}{2g}$. From the basic formula, § 82, we thus obtain the modified expression applicable to the pump :—

$$H_l - H_e - \frac{(v_2/2)^2}{2g}\left[1 - \left(\frac{r_1}{r_2}\right)^2\right] = 0{\cdot}5\,\frac{v_l^2}{2g} + \frac{4fl}{2b_l} \cdot \frac{v_l^2}{2g} + \frac{v_l^2}{2g}. \quad \textbf{(7-2)}$$

If only we had an accurate knowledge of the radial clearance b_l and of the friction coefficient f, then the velocity v_l of the leakage liquid could be computed from this expression, and the value of the leakage loss q_l would then be $(2\pi r_c b_l)v_l$. In regard to the radial clearance b_l, there is some doubt concerning it even when the pump is new, for the nominal dimension is so small—say 0·01 in. in a small pump—that manufacturing tolerances in machining the impeller and casing may materially influence its effective value. Then when the pump begins to work it is almost certain that the impeller will run slightly out of centre, Fig. 45 (i) ; and the resulting variation of b_l at different points around the circumference will again throw doubt upon its mean value. Still greater uncertainties surround the task of estimating the correct value of the friction coefficient f. The fundamental assumptions that are valid when assessing the value of the pipe coefficient f for a smooth circular pipe are obviously not valid now. In traversing the clearance space, the leaking liquid does not move parallel with the impeller axis but it has a helical or corkscrew motion. Nevertheless encouraging results have been reached by investigators who have tried to develop a relationship between Reynolds number and friction coefficient analogous to the well-known relationships applicable to circular pipes.

For present purposes, though, a much cruder conception will suffice : the clearance space may be regarded as an annular

orifice of cross-sectional area $\pi d_1 b_l$ through which the flow q_l may be evaluated by the formula

$$q_l = C_d \cdot (\pi d_1 b_l) \sqrt{2gH_l}$$

where H_l is the head-difference and C_d is a coefficient whose value may vary from about 0·3 to about 0·7.

Evidently in a single-inlet pump without balancing holes, Fig. 37 (i), leakage will take place through *one* clearance space only; but if the single-inlet impeller has a balancing chamber, Fig. 44 (ii), or if the impeller is of the double-inlet type, Fig. 37 (ii), then *two* clearance spaces will be available. Since additional back-flow will certainly occur through the water-sealed suction gland, § 85, the *gross* leakage loss from all causes, after the pump has been in service for some time, can only be estimated within very wide limits. As for "open" impellers,

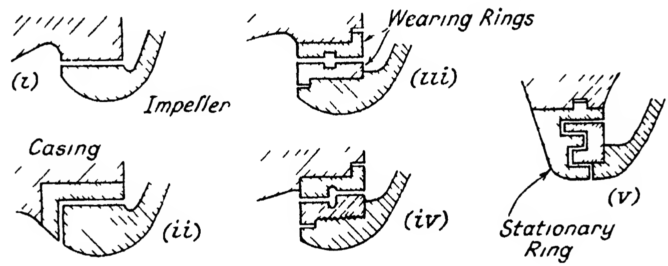


FIG. 48.—Types of sealing rings.

Fig. 37 (iii) and (iv), these manifestly fall outside the range of the above treatment. **(Example 11)**

84. Types of Sealing Ring. Some common ways of putting into practice the principles illustrated in Fig. 47 are shown in Fig. 48. Type (i) is of the basic pattern already seen in Fig. 47 (iii) which depends solely on hydraulic friction and on turbulence at inlet and at exit. This means that the radial clearance b_l must be kept down to its limiting value, which for small pumps is 0·01 in. (nominal). In type (ii), additional eddying is generated at the sharp bend in the clearance passage, and there is also a greater surface exposed to friction. Type (iii) relies upon grooves for increasing the hydraulic resistance, while the step in type (iv) directly recalls the joggled pipe in Fig. 47 (ii). A still more tortuous passage is contrived in type

(v), as a result of which the relative leakage loss, for a given radial clearance, may only be about one half of what it is in an equivalent plain cylindrical passage as at (i).

In point of fact the more complicated forms of passage are often preferred not so much to reduce the initial leakage loss as to discourage increased leakage during the pump's working life. By depending upon eddying at sharp corners for thwarting the tendency to back-flow, rather than upon surface friction, we can afford to give more generous radial clearances. Consequently if the impeller in course of service gains more lateral freedom, it will not be so likely to rub against the casing. Nor must one forget the abrasive particles. suspended in all but the purest filtered water, that are constantly trying to scour away the boundary surfaces of the clearance passages.

To allow for the wear that can clearly be foreseen, the pump designer makes the same provision that he does for other elements subject to wear: he provides renewable parts. Here they are called wearing-rings. During periodical overhauls of the pump, worn rings can quickly be stripped and replaced by new ones. Inexpensive pumps cannot afford such luxuries at all. Ordinary commercial pumps may have wearing rings in the fixed casing, while large high-head pumps are likely to have renewable rings both in the casing and on the impeller. Some methods of attachment are suggested in Fig. 48. The loose rings are sometimes screwed into place and sometimes pressed into place. It will be manifest that if the labyrinth pattern, type (v), is to be used for a split-casing pump, Fig. 39 (ii), then the stationary wearing rings must be slipped endwise into place on the impeller before the whole assembly is lowered home into the casing. They must then be located and locked before the top cover is put on.

In accordance with usual rules for renewable parts, fairly soft gunmetal or white-metal fixed rings would be suitable for the casing of pumps having unprotected cast-iron impellers, while bronze is indicated if both stationary and revolving wearing rings are specified.

85. The Stuffing-Boxes. At the points where the rotating shaft passes through the walls of the pump casing, packed glands usually serve to restrict leakage if suitable precautions are taken, § 81.

(i) *Under Positive Pressure.* In many types of pump the problem does not arise, because there are *no* glands exposed to positive pressure. All types of double-entry pumps belong to this category, Fig. 39, and so also do single-entry pumps having balancing holes and balancing chambers. Even where, as in Fig. 38 (i), the delivery liquid has access to the stuffing-box, its pressure is relieved by the forced-vortex motion described in § 74. Provided, therefore, that the gland-nuts are not unduly tightened, and that leakage liquid can be suitably disposed of, the system is unlikely to give trouble.

(ii) *Under Negative Pressure.* Just as some pumps have no *delivery* stuffing-boxes, so other pumps have no *suction* stuffing-boxes. Some examples are shown in Fig. 38 (i) and (ii). On the other hand, split-casing pumps have *two* suction stuffing-boxes, Fig. 39 (ii) and (iv). Leakage of air into the casing at these points cannot be tolerated; because of its mischievous possibilities, § 358, air must be excluded from the pump altogether. A *water-sealed gland* achieves this end simply and effectively, Fig. 49.

By means either of external pipes, Fig. 58, or by internal drilled or cored passages, Fig. 44 (ii), water from the pump volute is brought mid-way into the stuffing-box, where it is distributed by a *lantern-ring.* A zone of high-pressure water is thus established which atmospheric air is quite unable to pass. Leakage there will probably be- leakage of water *into* the suctions spaces of the pump, and leakage of water *out* into the atmosphere. But these are of little importance: indeed, because of their cooling and lubricating effect on the gland packing, leakages may actually be advantageous. To force home the gland in the hope of stopping the leakage may only bring about the troubles mentioned in § 81.

86. Stuffing-box Details. As for the packing rings themselves, the softer they are, the better. For cold-water pumps, it is usual to choose cotton impregnated with grease or with graphitic compounds. There may be packing-rings on both sides of the lantern-ring, Fig. 49 (ii), or on the outward (gland) side only, Fig. 49 (i). Pumps that have renewable wearing-rings will also have renewable neck-bushes at the inner ends of the stuffing-boxes, Fig. 49.

To facilitate the opening-up of split-casing pumps, both the glands and the lantern rings may be split horizontally. The designer will in any event attend to such obvious points as ensuring that sufficient axial space is left for withdrawing the gland and inserting the packing-rings.

Metallic seals in place of packed glands are sometimes preferred for relatively small pumps. A hardened revolving collar on the shaft is pressed by means of a spring against a corresponding hardened facing inside the casing.

87. The Casing: Inlet and Outlet Passages. This basic component of the pump has been left until the last because its construction must be adapted to so many purposes which first had to be explained, e.g. supporting the bearings, guiding the liquid, accommodating sealing devices, etc., etc. Referring to the functional classification in § 66, we arrive first at the consideration of the inlet and outlet passages. In all but the smallest pumps - say with openings of 2 in. diameter and less the suction and delivery branches may have standard flanges for mating with standard pipe flanges. But the size of the pump branches, and the diameter of the pipes in the main circuit served by the pump, need have no formal relationship one with the other. If they happen to differ, as they almost certainly will, then it is quite easy to interpose tapered reducing or enlarging pieces; indeed tapered enlargements help most effectively in the recuperation process, § 45 (ii).

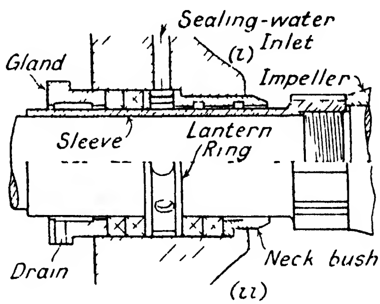


FIG. 49 Water sealed gland.

As a rule the pump suction branch is made a little larger in diameter than the delivery branch: thus a 4 to 5-in. pump has a 4-in. delivery branch and a 5-in. suction branch, and similarly for 8 to 10-in. and 12 to 14-in. pumps. The reason is to encourage the use of a generously-proportioned suction pipe which may be expected to impose only small energy losses on the incoming liquid; this in turn makes high suction lifts easier to attain, § 253. It is not feasible to work to predetermined limits of

velocity in the pump branches, because in a given pump working at its designed efficiency, the discharge will vary with the head, § 53. The notion of a flow ratio is preferable, using the notation that serves for establishing the impeller outlet area, § 52. If U_0 represents the mean velocity at the suction flange, and H_e the effective head on the pump, then the value $\frac{U_0}{\sqrt{2gH_e}}$ will vary from about 0·15 to 0·3, according to the specific speed of the pump. The connection between the two variables resembles the relation between n_s and ψ as plotted in Fig. 54, which means that the inlet area of the pump casing is much about the same as the outlet area of the impeller. As suggested above, the area at the delivery branch is usually, but not necessarily, 20 to 40 per cent. smaller than the suction area. According to these rules, the velocity at the branches may vary within the range 6 to 30 ft./sec.

In spite of this uncertainty, engineers find it convenient to use the diameter of the branches as a rough measure of the capacity of the pump ; they associate certain sizes with certain discharges, thus :—

Diameter of suction branch (inches)	*Rough notion of discharge (gallons per minute)*
4	200
6	700
12	3,000
24	11,000
36	25,000

88. The Volute. (i) *Hydraulic.* In § 45 the principles were explained which enabled the volute to serve as a recuperator of pressure-head. It is common practice to compute the cross-sections at various points around the perimeter in such a way that the mean velocity v_w is held more or less uniform, at a value equal to about 60 to 70 per cent. of the true velocity of whirl V_n of the liquid leaving the impeller, § 93. From the end of the volute passage the liquid is conveyed to the delivery flange through a conical or flared outlet passage of circular cross-section. But it is by no means essential that the *volute* cross-section should be circular ; examples of non-circular sections shown in Figs. 50, 51 and 78. Indeed, the

greater the radial width of the volute in relation to the axial width– the more the cross section is elongated– the greater is the opportunity for the helpful free-vortex motion depicted in Fig. 30 (ii). The offset volute shown in Fig. 51 (iii) has some of the advantages as a recuperator of the special type seen in Fig. 65. **(Example 6)**

(ii) *Mechanical.* In a standard type of volute the metal is not well disposed for resisting the bursting action of the liquid. As Fig. 50 (i) shows, the internal pressure, which is roughly equivalent to the manometric delivery head, H_{md}, tends to crack open the casing along a circumferential zone of weakness. In small pumps, and in medium-sized pumps destined only for

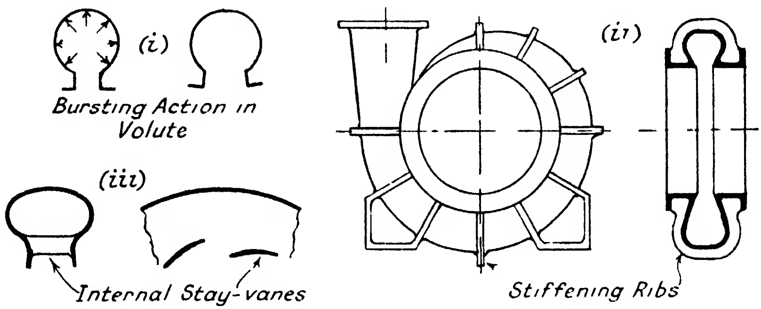


FIG. 50—Methods of stiffening casing.

low heads, no special provision is required to resist this tendency; but if the pressure is high or if the casing is large, reinforcement of some kind is essential. The first possibility is manifestly to cast the volute in steel instead of in iron. Constructional changes are shown in Fig. 50. The casing (ii) has external stiffening ribs: the casing (iii) has internal stay-vanes which serve as stream-lined tie-bolts. The internal reinforcement makes a neater job, but there is evidently a risk that the stay-vanes may interfere with the flow of the liquid in reduced-flow and increased-flow conditions, Chapter XIV.

89. The Casing: Other Details. The typical forms of construction illustrated in Figs. 38 to 40 have suggested how manifold are the methods of supporting the volute or of using it to serve as a support for other components. In regard to general design, it is only necessary here to mention again the

requirements listed in § 68. Rigidity is a prime essential. Some other points are :

(i) *Inlet Volute.* In only one disposition of suction branch, that shown in Figs. 38 (i) and 40 (ii) in which the liquid approaches the side-inlet impeller axially, are the inlet conditions beyond reproach. All other forms embody a bend or elbow which imposes irregular velocity components on the liquid just at the time when we should prefer complete uniformity. The passages of the normal split-casing pump, Fig. 39 (ii), look as though they would be particularly liable to dissipate energy and thus to lessen the suction capabilities of the pump, § 253. Some makers of such pumps therefore provide very generous inlet passages, and by fashioning them into semi-volutes they

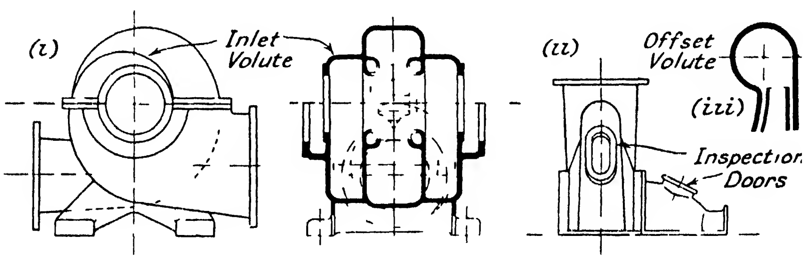


FIG. 51 —Other details of casings

give the liquid just the guidance it likes. An example is shown in Fig. 51 (i). If the liquid at the moment of entering the impeller is found to have kept some of the whirl velocity deliberately forced upon it, this will not impair the pump performance, so long as the impeller blade angles are suitably modified.

(ii) *Inspection Openings.* Here is another reference to § 68. Although the designer of the casing may always keep accessibility in mind, other considerations may take precedence. So if he cannot promise easy dismantling of the pump, he can at least offer ease of inspection on a limited scale. Hand holes in the suction elbows, closed by bolted doors, are extremely convenient, Fig. 51 (ii). The reason why attendants usually want to look inside the pump here, is to remove rubbish that may have wrapped itself round the shaft or partially choked the impeller openings ; and by means of the inspection openings they can do this quite quickly. In very large pumps it may

also be worth while to provide a man-hole on the volute itself, Fig. 51 (ii).

Centrifugal Pump Design

90. Routine Computations. The intention in this part of Chapter VII is to lay down a routine for establishing as quickly as possible the main dimensions of the types of pump that have been described in earlier paragraphs. The justification for the general trend of the information will be found in the chapters on Performance in Part C of the book, and especially in Chapter XIII.

In accordance with the general programme proposed in Chapter VI, the first step in design is to put the *basic data* in the form :-

H_e effective head in feet § 163,

Q discharge in cubic feet per second.

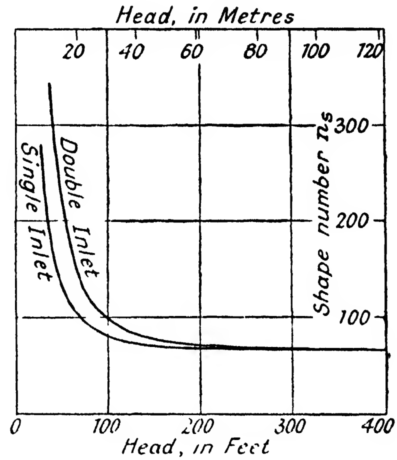


FIG. 52 Relationship between head and limiting shape number.

Next the *shape number* (specific speed) is estimated. If no other guidance is available, the appropriate shape number of impeller to suit the specified head can be read off directly from the graphs, Fig. 52. They are intended to indicate a *general trend only*, and not to lay down a rigid correlation. Thus small low-head commercial pumps often have shape numbers much lower than those suggested.

91. Speed, Power, Efficiency. From the selected shape number n_s (or from the specific speed), the

Rotational speed N in revs. per min. is extracted from formula (5-4), § 59 :—

$$n_s = \frac{\frac{1000\,N}{60}\sqrt{Q}}{(gH_e)^{\frac{3}{4}}}$$

A more convenient form, *suited to the units : feet, cub. ft./sec.*, is :—

$$n_s = 1{\cdot}232\,\frac{N\sqrt{Q}}{H_e^{\frac{3}{4}}}.$$

The *Water Horse-power*, or power output, has the value

$$\text{W.H.P.} = \frac{QwH_e}{550}$$

where w = density of liquid in lbs./cu. ft.
= 62·3 for clean water at normal temperatures.

Gross or Overall Efficiency, η_m. This varies with the output power and the shape number in the general manner shown in the graphs, Fig. 53. A suitable value is found by interpolation. Again it must be remembered that such graphs only purport to show trends or tendencies (*). The type of pump – whether split-casing, end-suction, etc., etc. – may influence the efficiency in a manner that a general graph cannot represent, § 195.

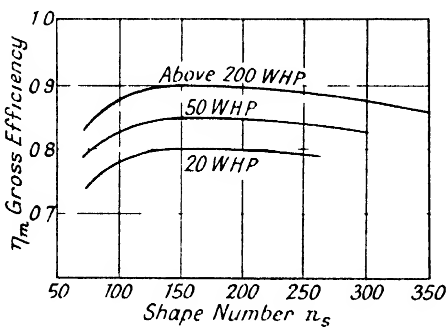


FIG. 53.—Relationship between shape number, output and efficiency.

Moreover, Fig. 53 relates only to the performance of *new pumps under the most favourable conditions*. The probable *service* efficiency of the pumps may be a few per cent. lower, § 258.

Shaft Horse-power, S.H.P. This is the input power to the pump shaft. Its value is extracted from the relationship :—

$$\text{Gross efficiency} \quad \eta_m = \frac{\text{W.H.P.}}{\text{S.H.P.}} \quad . \quad . \quad (\S\ 165)$$

By adding a reasonable allowance to the shaft horse-power, we arrive at the requisite B.H.P. of the motive unit, § 276.

Charts and graphs facing p. 480 will expedite some of the above computations.

92. Impeller Dimensions. The *shape ratios* defined in § 52 are useful here. Their average values as used in practice are linked in a general way with the impeller shape number, thus :—

Having decided upon the type of rotor, whether single-inlet or double-inlet, § 69, we find the *speed ratio* ϕ from the graph, Fig. 54. Then by means of the expression $\phi = \frac{v_2}{\sqrt{2gH_e}}$, the value of the impeller rim velocity v_2 in feet per second is derived.

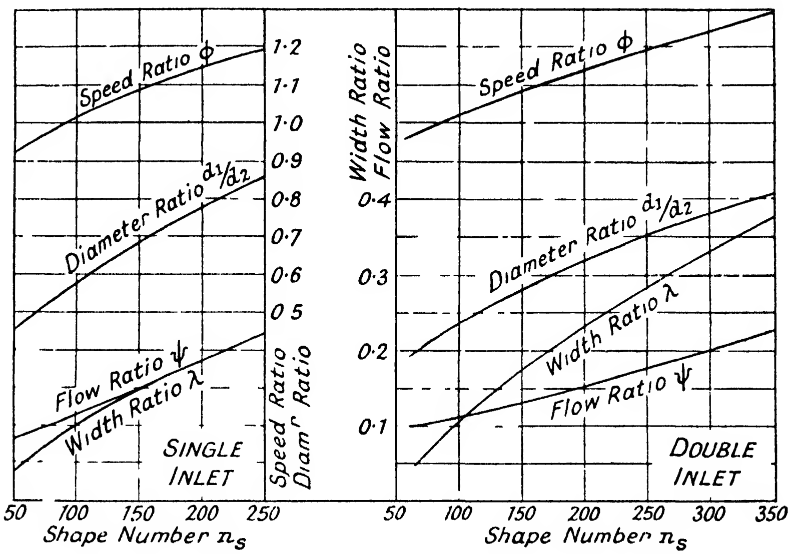


FIG. 54. Relationship between shape number and shape ratios.

Outside Diameter, d_2. To obtain this, known values are inserted in the equation $v_2 = \frac{\pi d_2 N}{60}$.

Width at Outer Periphery, b_2. This axial distance between the shrouds is found by first estimating from Fig. 54 the proper value of the width ratio λ, and extracting from the equation $b_2 = \lambda d_2$ the desired width. This first provisional estimate must afterwards be corrected as in § 94.

Inner Diameter, d_1. The ratio d_1/d_2 varies not only with the shape number, Fig. 54, but also upon the type of impeller.

Inner Width, b_1. This dimension is often so chosen as to fulfil the condition $b_1 d_1 = b_2 d_2$; but sometimes it has a smaller

value, i.e., the velocity of flow *decreases* as the liquid flows outwards. In any event a subsequent correction is essential, § 94.

Number of Impeller Blades, n. This varies from a minimum of six for very low specific speeds, to a maximum of about twelve for very high specific speeds.

Thickness of Impeller Blades, t. This will naturally depend upon the size of the wheel, the kind of metal, and the skill of the founder. Thin blades are favourable to good initial hydraulic performance, but they naturally cannot withstand cavitation erosion, (§ 257), as well as thicker blades can. A likely value of t for cast iron blades may range from 4 to 8 mm.

The general proportions of impellers shaped according to the foregoing rules are illustrated in Fig. 55.

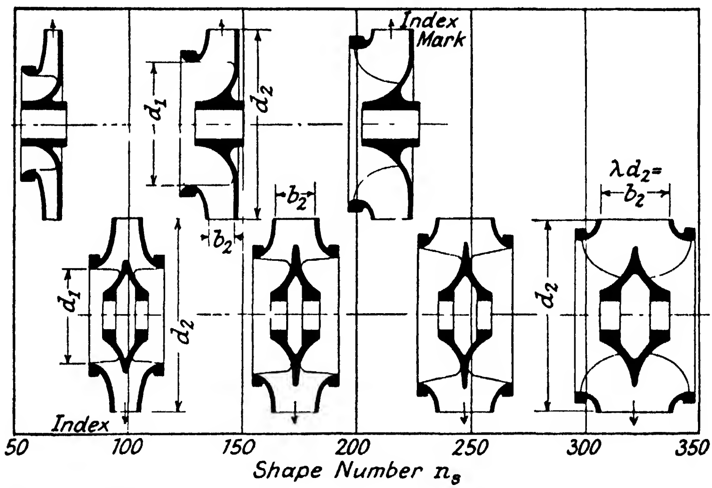


FIG. 55.—Relationship between shape number and impeller proportions. (Read the index mark against the n_s scale.)

93. Impeller Blade Angles ("Cylindrical" Blades).

This paragraph deals only with blades suited for two-dimensional flow ; they are described as cylindrical because they form part of cylindrical surfaces—though the cylinders are not necessarily circular cylinders. The blade surfaces could be generated by a line maintained always parallel with the impeller axis. Such rotors are adapted only for low and medium specific speeds. Mixed-flow blading (§ 28) is described in §§ 99, 107.

The routine of calculation proceeds thus :—

Hydraulic Efficiency, η_h. This value, § 164, is always greater than that of the gross efficiency, η_m, and may be tentatively estimated from the relationship :—

$$\frac{1 - \eta_h}{1 - \eta_m} = K_h$$

where K_h ranges from about 0·5 for low specific speeds to about 0·8 for high specific speeds.

True Whirl Component, V_n. In order that the pump may generate the stipulated head H_e, the liquid must leave the impeller with a tangential velocity component V_n that can be computed from the relationship

$$H_e = \eta_h \cdot \frac{V_n v_2}{g} \quad . \; . \; . \; \S\ 164.$$

Ideal Whirl Component, V_∞. Because the response of the liquid to the tangential impulsion of the blades is imperfect, § 17, the outlet impeller blade angle must be suited to an ideal tangential component V_∞ that is *greater* than the true one. The relation between the two is highly complex, and only a rough approximation can now be offered. It is :—

$$V_\infty = V_n\left(1 + \frac{K_n}{n}\right).$$

where n is the number of blades, and K_n is a factor that can be taken as about 3 for *low* specific speeds, and as about 5 for *high* specific speeds.

Velocity of Flow. Having extracted from the appropriate graph, Fig. 54, the value of the *flow ratio* ψ, we can at once compute the value of the outlet radial flow component or velocity of flow Y_2 from the expression $Y_2 = \psi\sqrt{2gH_e}$.

Outlet Blade Angle, γ. This is found from the expression

$$\cot \gamma = \frac{v_2 - V_\infty}{Y_2}.$$

Inlet Blade Angle, β. Assuming that the inner edges of the blade are set on a circle of the same diameter d_1 as that of the impeller eye, the value of the inlet flow velocity Y_1 is extracted thus :—

$$Y_1 = Y_2 \cdot \frac{d_2 b_2}{d_1 b_1}.$$

Since the peripheral velocity v_1 at diameter d_1 is $\frac{\pi d_1 N}{60}$, the value of the inlet blade angle can be found from the expression

$$\tan \beta = \frac{Y_1}{v_1}.$$

Velocity Triangles. By plotting the velocity triangles as in Fig. 56, a graphic impression of the general solution of the problem can be formed.

94. Corrections to Impeller Width. Two factors still remain to be accounted for. They are (i) the internal leakage

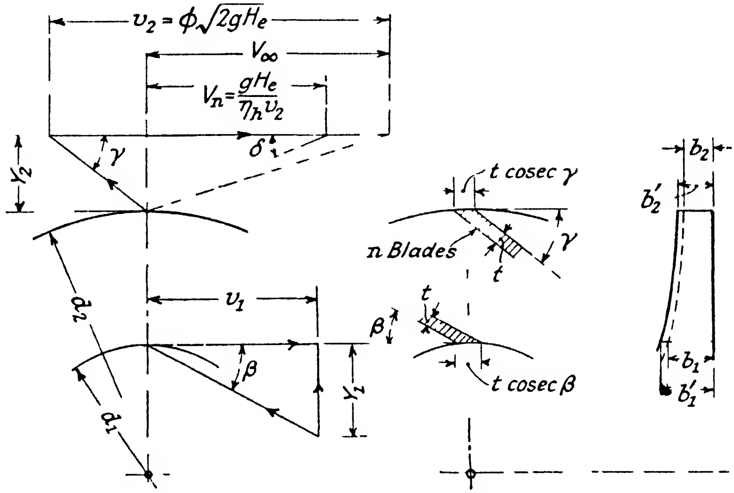


FIG. 56 – Construction for establishing impeller blade angles, etc.

loss q_l in the pump, § 82, and (ii) the obstruction to flow offered by the thickness of the metal forming the n blades. A single correction will take care of them both: all that need be done is to move the impeller shrouds or discs a little further apart. If, in the diagrammatic cross-section in Fig. 56, the broken line indicates the provisional position of the impeller shroud as computed in § 92, then the full line will show the final position. This slight shift will have no adverse effect, for the values of the width ratio plotted in Fig. 54 are nominal only, based on the simplified conditions hitherto in operation.

According to §§ 82 and 83, the leakage loss q_l may amount to anything from 0·03 Q to 0·10 Q, where Q is the net quantity

coming out of the delivery pipe; and naturally in framing an estimate for a particular design we shall try to imagine not what the leakage will be when the pump is new but when it has had a long period of service, § 258. Also it is evident from Fig. 56 that the effect of the blade thickness t is to reduce the effective impeller inlet area from $\pi d_1 b_1$ to $(\pi d_1 - nt \operatorname{cosec} \beta) b_1$, and to reduce the effective outlet area from $\pi d_2 b_2$ to $(\pi d_2 - nt \operatorname{cosec} \gamma) b_2$.

Consequently the corrected inlet impeller width will have the value

$$b'_1 = b_1 \cdot \left(\frac{Q + q_l}{Q}\right) \cdot \frac{\pi d_1}{\pi d_1 - nt \operatorname{cosec} \beta}$$

and the corrected outlet width will have the value

$$b'_2 = b_2 \left(\frac{Q + q_l}{Q}\right) \cdot \frac{\pi d_2}{\pi d_2 - nt \operatorname{cosec} \gamma}.$$

As a final check, one could compute directly the values :—

$$b'_1 = \text{inlet width} = \frac{Q + q_l}{Y_1(\pi d_1 - nt \operatorname{cosec} \beta)}.$$

$$b'_2 = \text{outlet width} = \frac{Q + q_l}{Y_2(\pi d_2 - nt \operatorname{cosec} \gamma)}.$$

Similarly, after the shaft diameter has been estimated and the minimum diameter d_{b0} of the impeller boss is known, the net inlet area of the impeller eye can be worked out; this should not be materially less than the net inlet area between the blades. That is to say: for a single inlet impeller

$$(\pi/4)(d_1^2 - d_{b0}^2)$$

should not be less than $(\pi d_1 - nt \operatorname{cosec} \beta) b'_1$.

95. Impeller Blade Shape. An infinite number of blade shapes might be struck, each having the specified initial angle β and terminal angle γ. For ordinary purposes only two of these are worth considering, a compound curve formed of two circular arcs, and a simple curve formed of a single circular arc. The methods of construction are illustrated in Fig. 57 (I) and (II) and explained below.

Compound Curve (I). The successive steps are :—

(i) On the circle representing the inner impeller rim, mark off points A and B spaced apart according to the number of blades n.

(ii) Set off line AC making an angle β (the inlet blade angle), with radius OA.

(iii) Draw the base circle of radius OD tangent to line AC.

(iv) Draw DB tangent to the base circle, and continue the line beyond A_1.

(v) On line DBA_1 mark off as shown the distances t, l_1, and t, where t is the blade thickness and l_1 has the value $\dfrac{2\pi(\overline{OD})}{n}$.

(vi) Choose a point C on line AC such that, with C as centre, an arc of radius r_3 will pass through A and A_1.

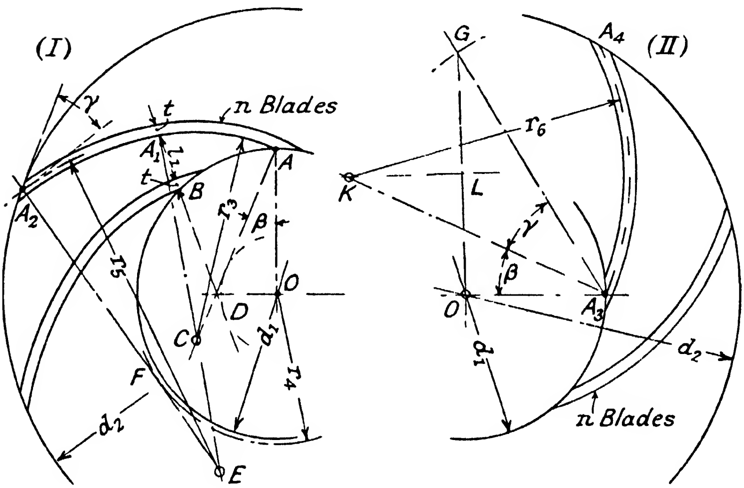


FIG. 57. Construction for setting out blade shape.

(vii) Draw the first part of the blade, as shown, with radii r_3 and $(r_3 + t)$ respectively.

(viii) On line A_1C produced, choose a point E as centre such that arc A_1A_2 may be struck with radius r_5, to fulfil the condition that the outlet blade angle γ has the specified value. This must be done by trial and error, but the work may be simplified thus: With O as centre, strike another construction circle of radius $r_4 = \frac{1}{2}d_2 \sin\gamma$. Draw line EFA_2 touching the circle at F. If then distance EA_1 = distance EA_2, the line can stand; if not, try again.

(ix) With point E as centre, draw the arcs forming the second part of the blade.

Simple Curve (II). The steps are :—

(i) From point A_3 on the inner impeller rim, set off the angles $OA_3K = \beta$ and $KA_3G = \gamma$, as shown in Fig. 57 (II).

(ii) Mark off distance A_3G = outer impeller radius $= d_2/2$.

(iii) Draw KL intersecting OG at right angles.

(iv) With centre at K, viz. at the intersection of KL and A_3K, and with radius r_6, draw the arc A_3A_4 representing the centre line of the blade.

Comparison with Figs. 6 and 7 shows that the compound curve I resembles the ideal blade form more closely than curve II does.

96. Production Routine. The degree of *finish* given to the rotor casting has an appreciable effect on the pump performance and efficiency. Iron castings as a rule are only machined on the surfaces where this is essential ; elsewhere the skin is smoothed by hand. But when bronze castings are used, and when it is desired to maintain the highest possible efficiency over the longest possible period, then the whole of the external surfaces should be machined and polished, and the internal surfaces hand-finished and polished as far as this can be done.

Then the impeller must be *balanced*, first perhaps on a temporary mandrel and afterwards when finally mounted on its shaft. Metal can be removed by filing or machining, until in the end, when the shaft is set on levelled straightedges or on a balancing machine, the assembly remains in any position.

The *casing*, designed in accordance with the rules of § 88, will have to pass a hydraulic pressure test before the pump is assembled.

Methods of conducting *running tests* on the finished pump are described in Chapter XII. **(Example 12)**

97. Standard Designs and Modified Designs. An example of a small pump designed by the methods prescribed in §§ 90 to 95 is illustrated in Fig. 58. But although the graphs on which the design is based (Fig. 54) are representative of current practice, that does not mean that successful pumps can only be built by conforming to the stipulated values of speed

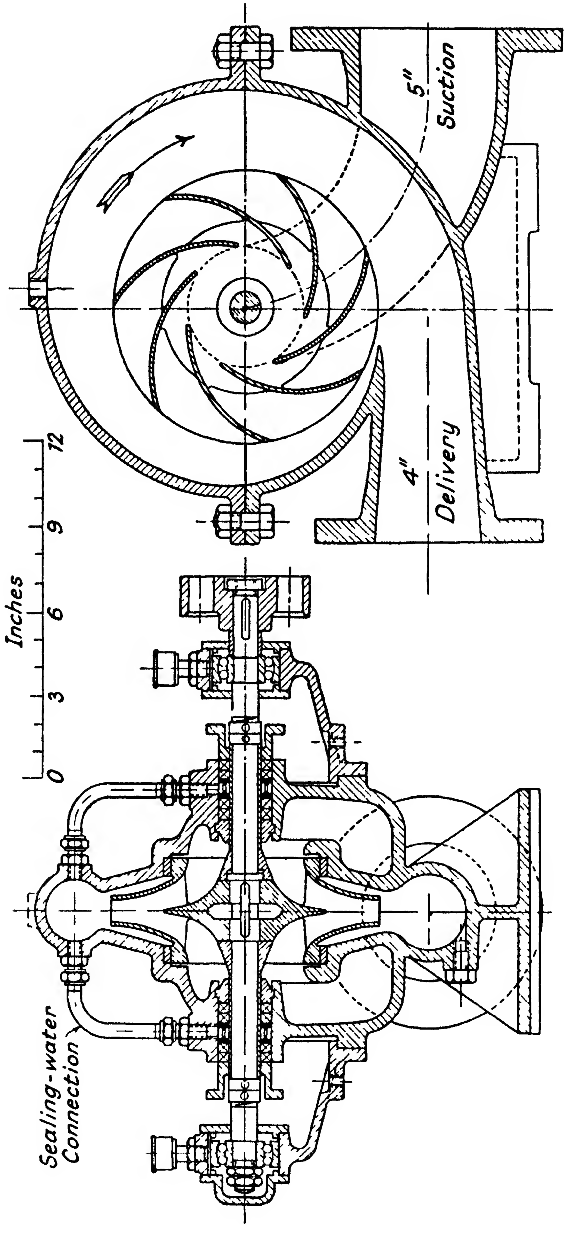


FIG. 58.—General proportions of split-casing centrifugal pump to suit conditions:—

Effective head = 60 feet.

Discharge = 400 gallons per minute.

Speed = 1450 r.p.m.

ratio, flow ratio, and the like. On the contrary, it may be necessary to make wide departures from these average values. By means of such deviations the *characteristic performance* of the pump may be modified within certain limits, § 214, and so also may the *suction-lift* capacity of the pump, § 253. Nor need these changes necessarily alter the numerical value of the specific speed or of the shape number; so long as the product $\phi\sqrt{\lambda\psi}$ remains the same, so also will the shape number remain unaltered, § 59. It is important to remember this, otherwise one might get the impression that the geometrical shapes illustrated in Fig. 55 are the only shapes that can be associated with a given shape number. They are not. They are only the shapes that experience has shown most likely to be successful in ordinary circumstances.

Another departure from standard practice is the use of a diffuser-ring type of recuperator, § 44, instead of the volute recuperator. As this practice is finding less and less favour for single-stage pumps, it is discussed in this book only in connection with multi-stage pumps, § 122.

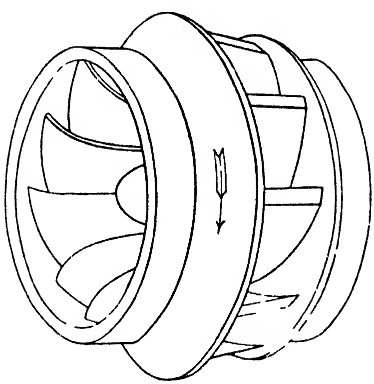


FIG. 59. Mixed-flow centrifugal pump impeller. (See § 99.)

CHAPTER VIII

MIXED-FLOW AND AXIAL-FLOW PUMPS

98. Range of Types. Continuing to use the terminology of Chapters III and IV (but remembering that only in this book are these terms consistently used), we can say that the present chapter concerns itself with :—

(i) Mixed-flow centrifugal pumps.
(ii) Screw pumps.
(iii) Half-axial pumps.
(iv) Axial-flow or propeller pumps.

In traversing the continuous range of types mentioned in § 4 and illustrated in Fig. 36, we meet with rotors of steadily increasing shape numbers or specific speeds. As the rotors progressively change their shape we shall find, too, that the axial velocity component becomes more and more predominant, § 28. From the point of view of duty or purpose, we have already seen from § 64 that the pumps now to be studied are specially suitable for raising large volumes of water against relatively low heads. To show that no sharp dividing line can be drawn between the machines described in Chapter VII and the mixed-flow pumps described in this chapter, it can be noted that the same diagram sometimes relates to both types. Thus the impellers shown on the left of Fig. 55 are suited to the centrifugal pumps of Chapter VII ; those to the right are mixed-flow impellers and their blade form is discussed here. But if some arbitrary limiting numeral is demanded, one can say that the following paragraphs all relate to pumps having a shape number greater than about 150 to 200.

Although in general the pumped liquid will still be cold water, it can often only be termed clean in relation to the particular kind of duty required. Thus the pump must be ready to accept such impurities as are common in river, sea, and canal water, e.g., suspended mud and silt, and such floating refuse as has passed through coarse grid screens. Sometimes sewage must be handled.

99. Mixed-flow Centrifugal Pumps. The general proportions of mixed-flow rotors, both of the single-entry and double-entry types, can be read from the appropriate parts of Figs. 54 and 55 ; it is only in regard to the blading that the descriptions given in Chapter VII require to be supplemented.

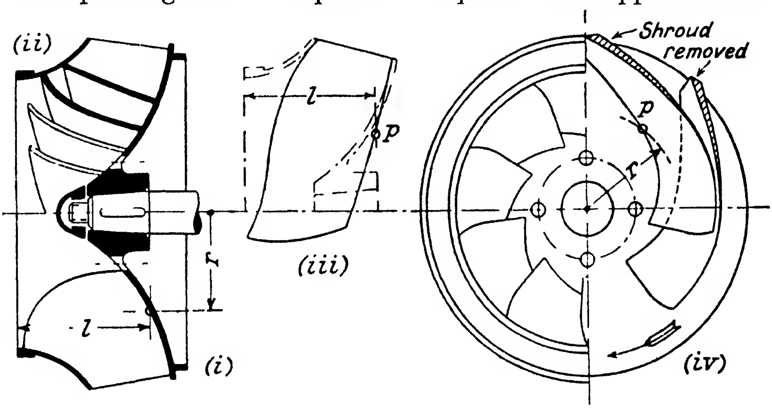


FIG. 60.—Conventional and true projections of mixed-flow impeller blades.

To distinguish mixed-flow single-inlet impellers from the (nominally) radial-flow type with so-called "cylindrical" blades, § 93, the mixed-flow form is sometimes given the name "Francis" because of its resemblance to a Francis turbine runner. Occasionally the term "helico-centrifugal" is used.

In developing along practical lines the elementary treatment of three-dimensional flow offered in §§ 32 to 34, we have now to think not so much about the liquid as about the patternmaker and the moulder and the machinist ; and the more so because of the complex forms that the blades must inevitably take (*). The first essential is to get a clear impression of what the blades look like. The perspective view of a mixed-flow impeller in Fig. 59 should help, while various forms of orthographic projection are given in Fig. 60. The orthodox system

(i) that has hitherto been used for all earlier diagrams is purely conventional; it does not represent the actual appearance of the blade, but only gives a projection on to a diametral or meridional plane, § 31. Suppose, for instance, that we wish to plot according to this system the position of the point p whose true position is shown in the other views in Fig. 60; then we truly mark off its radial distance r from the rotor axis and its axial distance l from the inlet plane, but we take no account of the angular position of the point in relation to a selected diameter. This system is advantageous when questions of blade loading are to be studied, § 26. Besides the true projections, Fig. 60 (iii) and (iv), which correctly depict the blade form, it is worth while to study the true cross-section (ii), for it gives a good impression of the shape of the passages through which the liquid flows.

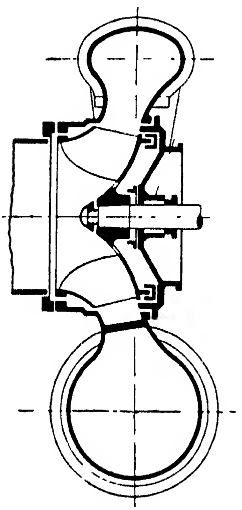


FIG. 61. Mixed-flow side-inlet centrifugal pump.

The volute type of *recuperator* invariably used for mixed-flow centrifugal pumps only differs in its proportions from other volutes, § 45. The example shown in Fig. 61 is suited to the impeller depicted in Figs. 59 and 60; the assembly relates to the vertical-shaft pump shown in Fig. 40 (ii), but is here shown turned on its side for convenience of comparison with other mixed-flow pumps. The volute for a double-entry mixed-flow pump has been shown in Fig. 51 (i).

100. Screw-type Rotors. Hydraulically there seems to be little difference between the screw-type rotor seen in Fig. 62 and the Francis type impeller shown in Fig. 59. Mechanically there is one very real distinction; whereas the impeller has an outer ring or shroud for supporting the blades, the screw-rotor has not. So long as the shroud was present we have felt justified in calling the pump a centrifugal pump; when it has disappeared we have to use the names *screw pump* or *half-axial pump*, § 47. In this book the term screw pump implies a volute type of recuperator, while we associate the name half-axial with

a co-axial recuperator. The screw-type rotors for the two categories are virtually identical (*).

The corresponding views in Figs. 60 and 63 suggest that the individual blades of the screw-rotor are not dissimilar from

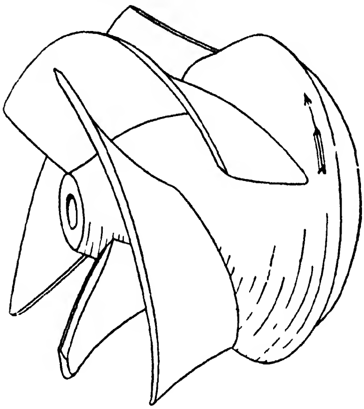


FIG. 62. Screw-type rotor.

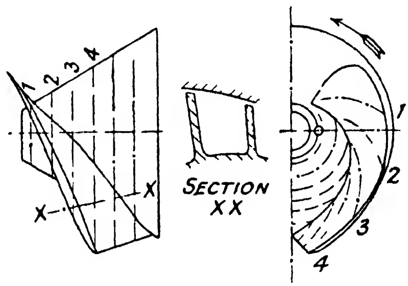


FIG. 63.—True shape of screw-type rotor blades.

those of the mixed-flow impeller; but whereas the impeller may have ten or twelve blades, § 92, the screw-rotor only has

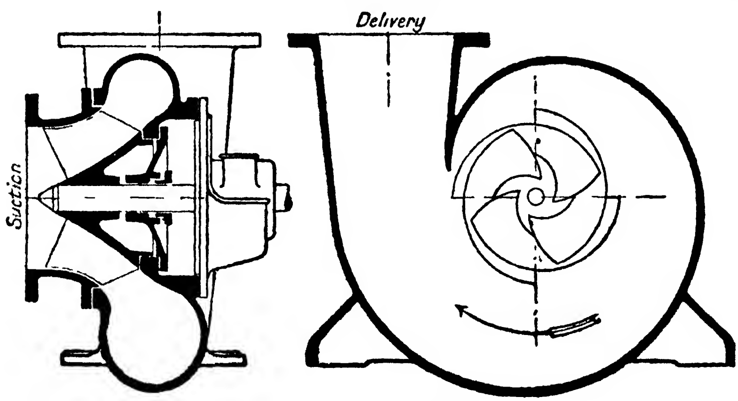


FIG. 64. Screw pump.

three or four. Although the blades of the screw-rotor are so few, yet the spaces between them may still properly be called passages, as can be seen from the section XX in Fig. 63.

101. Screw Pumps. From the example of a screw pump illustrated in Fig. 64, it is evident that there is still a basic

resemblance to the original side-inlet centrifugal pump, Fig. 38 (i). In the alternative design in which the rotor is supported by a bearing on each side, Fig. 65, a recuperator is shown which

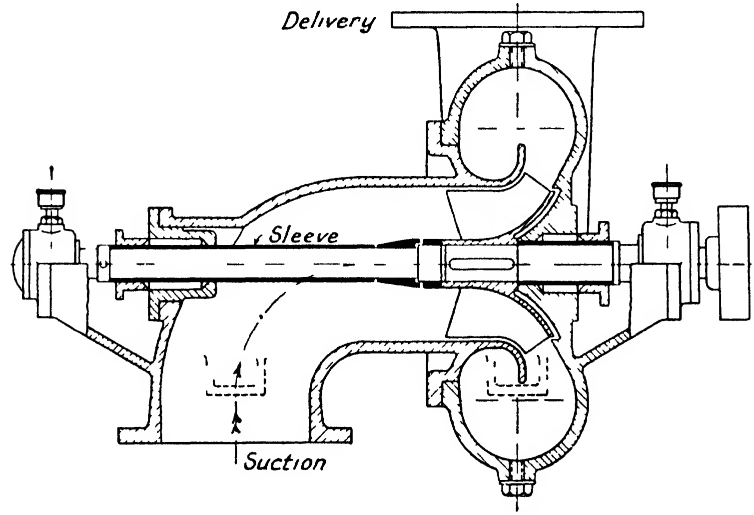


FIG 65.—Alternative form of screw pump

is not a true volute; the recuperator is rather intended to encourage a circulation of the liquid in a direction at right angles to the pump axis. To borrow an illustration which was first

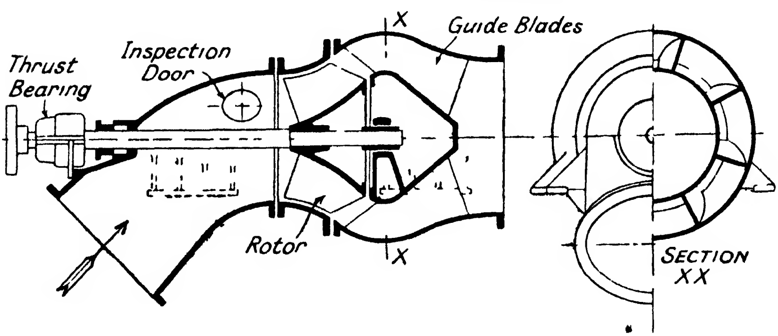


FIG. 66.—Half axial pump.

applied to the hydraulic coupling, one might say that the path of an element of liquid after leaving the rotor resembles "the stripe on a barber's pole that has been bent into a circle".

This type of recuperator is found to be quite effective, alike for screw-rotors and for mixed-flow centrifugal impellers.

Screw pumps may be arranged either with *horizontal* or with *vertical* shafts. (*).

102. Half-axial Pump. The combination of a screw-rotor and co-axial recuperator, Fig. 66, is nearly always arranged with horizontal shaft. From the perspective view of the recuperator, Fig. 67, an impression may be formed of the way in which the inlet edges of the blades are canted over to suit the direction in which the liquid leaves the rotor. The illustrations make it clear that the number of these guide blades is always *greater* than the number of rotor blades.

Since the performance and efficiency of screw pumps and half-axial pumps are so nearly equal, the choice between them is often governed by their adaptability to the particular lay-out of the pipes and conduits in which the pump is to be interposed.

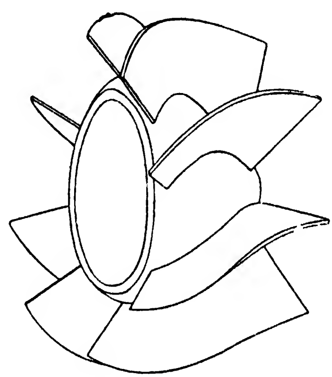


FIG. 67.—Form of recuperator blades for half-axial pump.

103. Axial-flow or Propeller Pumps. Two features common to all types of propeller pump are (i) the general direction of flow through the pump is (or is assumed to be) purely axial : there are no radial components, § 35 ; and (ii) the recuperator is invariably of the co-axial form of the general type shown in Fig. 32 and Fig. 67. No violent transition is therefore necessary to bring us to the extreme or limiting type of rotodynamic pump ; to convert the (nearly) axial direction of flow in the half-axial pump to the fully axial flow of the propeller pump, we need hardly do more than replace the screw-type rotor by a propeller-type rotor.

Axial-flow pumps may be arranged with (*a*) *horizontal*, (*b*) *inclined*, or (*c*) *vertical* shaft. Their rotors may have blades set at an invariable angle, or the blades may be adjustable ; this possibility of adjustment forms a valuable attribute of these pumps.

The vertical unit depicted in Fig. 68 gives a good impression

of the extremely compact disposition of such pumps In addition to the essential rotating propeller blades and fixed recuperator blades, there may also be fixed radial inlet guide-blades as shown by broken lines in the diagram ; their purpose is to impose true axial flow on the water, i.e., to remove any accidental whirl components, before the water enters the zone of influence of the rotating blades. Although the rotating blades and the boss or hub which supports them may be cast in one piece, it is often convenient to form the blades separately even if no adjustment is envisaged while the pump is running. Possible methods of attachment are

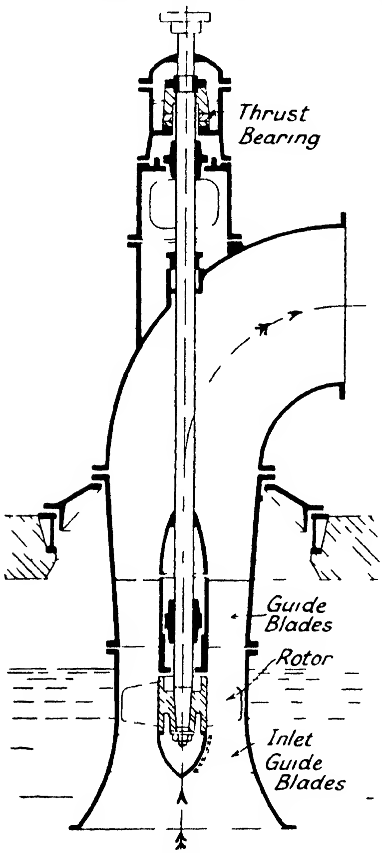


FIG. 68 Vertical shaft propeller pump

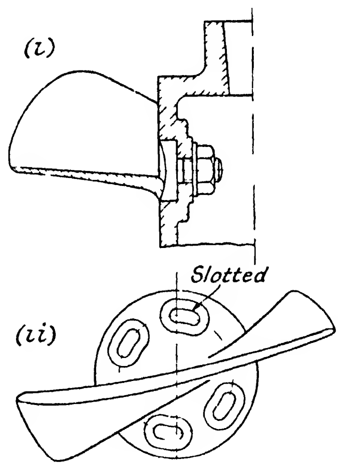


FIG. 69 Attachment of propeller blades

indicated in Fig. 69, suited for (i) relatively small pumps and (ii) relatively large pumps ; the limited range of adjustment of blade angle is usually sufficient to permit performance to be suitably corrected after the pump has been tested, drained, and opened up.

104. Pumps with Variable-pitch Propellers. If the

range of variation of the blade angles is considerable, thus permitting the pump performance to be controlled at will during normal service, then the axial-flow rotor is described as a variable-pitch propeller. The mechanical basic resemblance to a variable-pitch airscrew is very close. The principle as applied to an inclined-shaft pump is illustrated in Fig. 70. Each of the blades—which here are four in number—has a stub-shaft which projects into the interior of the hollow hub

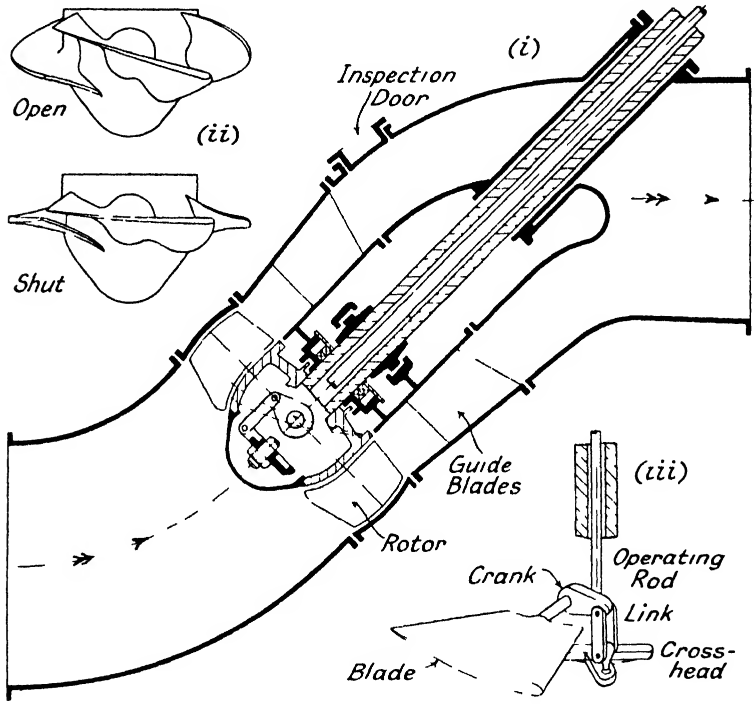


FIG 70 Inclined shaft variable pitch axial flow pump

or boss rather after the fashion of Fig. 69 (i). But now proper provision is made for lubricating the stub-shaft as it turns; and for communicating the turning movement there is a short crank actuated by links from a cross-head attached to an axial operating rod. From the small key diagram (iii) and from the sectional view (i), Fig 70, it is clear that movement of the operating rod which passes through the hollow pump-shaft will cause each of the four blades to swivel about its axis and

thereby alter the inlet and outlet angles. Diagram (ii) shows the appearance of the rotor with blades fully opened and with blades fully closed. The very beneficial effect on the pump performance of pitch variation is explained in § 216 (*b*).

If the casing in the region of the rotor were truly cylindrical, it would not be possible to maintain, for all blade positions, the desired small radial clearance between the outer rim of the blades and the walls of the casing. Fig. 70 (i) explains how this difficulty is overcome. So far as possible, the appropriate surfaces are made of spherical form, struck from a centre lying at the intersection of the stub-shaft axis with the main pump axis.

As for the manner of adjusting the blade angles, there are two possibilities: (i) after momentarily stopping the pump, a pinion may be brought into mesh with a rack on the operating-rod, and an external handwheel used to move the rod, (ii) if the operating rod can be controlled by an external revolving sleeve, then the pitch of the blades can be altered while the pump is running. Some type of servo-motor attachment will make remote control feasible.

Mechanical Details and Working Proportions

105. Details of Construction. Many of the factors which govern the design and construction of centrifugal pumps, Chapter VII, are operative here also, so it will suffice now to indicate the main departures from the descriptions already given.

Casing. As delivery pressures are always relatively low, there is little difficulty in designing the casing so that it has the necessary mechanical strength. The casing is often so big that it is necessarily bolted together from separate castings, with one of the main joints lying preferably in the diametral plane. Brackets cast on the lower sections transfer the weight of the pump to concrete foundations or to cast or rolled beams.

Axial Thrust. Except in mixed-flow double-inlet centrifugal pumps, there is no possibility of eliminating, even ideally, the resultant hydraulic axial thrust on the rotor. It is true that balancing holes, § 77 (ii), will considerably reduce the thrust on other types of mixed-flow rotors, and these holes are

in fact indicated in Figs. 60 and 63. No relief at all can be found from the still greater axial thrust on axial-flow rotors, a thrust whose value can be estimated by the methods of §§ 37, 113 (iii). Consequently the thrust-bearing, § 78, nearly always requires serious thought.

Bearings, Shaft, Stuffing-box, etc. The heavily-loaded thrust bearing just mentioned may be of the ball type for small pumps, but for large pumps there is no alternative to the tilting-pad type. Air-cooled Michell bearings are provided for the pumps in Figs. 66 and 68; artificial cooling is often necessary for larger units, either by water-jacketing the bearing housing or by circulating the oil through a cooler. Internal *journal or steady bearings*, as in Figs. 66 and 68, may be white-metal lined and lubricated by grease forced through small-bore piping by an external grease-pump. Bearings lined with water-lubricated "plastic" or with self-lubricating bronze are also available.

The shaft *sleeves*, § 80, often protect the whole surface of the shaft that is exposed to the liquid; an example is seen in Fig. 65. Monel metal is a good material for them.

If *water-sealed stuffing-boxes* are necessary, § 85, they may need a special supply of filtered water, for the muddy or dirty water passing through the pump would do more harm than good if it were fed to the lantern-ring. If the pump works in a siphonic circuit, Figs. 200, 201, 204 (I), an independent supply of sealing-water under pressure is likewise essential.

Sealing-rings, etc. In mixed-flow centrifugal pumps, internal leakage may be controlled by sealing-rings, renewable or otherwise, as in § 83. Sealing-rings may likewise reduce the leakage loss through the balancing holes of screw-type rotors. But the leakage past the unsupported edges of screw-type rotors and axial-flow rotors can be kept within permissible limits in one way only, viz. by maintaining the smallest practicable running clearance between the blade edges and the internal walls of the casing. Sometimes the casing in this zone has renewable or adjustable sections.

106. Design Data for Screw-type Pumps. These notes are supplementary to those in §§ 90 to 96 that relate to centrifugal pumps. It must here be pointed out that the relation between shape number and working proportions no longer follows such a regular progression as was plotted in Fig. 54.

Shape Number. From § 64 we find that the shape number of screw-type rotors should be kept within the range 250 to 500. By the method of § 59, it can be shown that *if nominal values* are used as explained below, the shape number can be put in the form $n_s = 475\phi\sqrt{\psi}$.

Gross Efficiency. Although in a very general way the graphs reproduced in Fig. 53 give some sort of guidance, at least as regards trends and tendencies, it is difficult to give any more precise statement than this, that even for screw-type pumps exceeding 200 h.p., the gross efficiency η_m rarely exceeds about 0·87.

Rotor Proportions. It is convenient to use nominal values of the speed ratio ϕ and the flow ratio ψ, based on the *inlet*

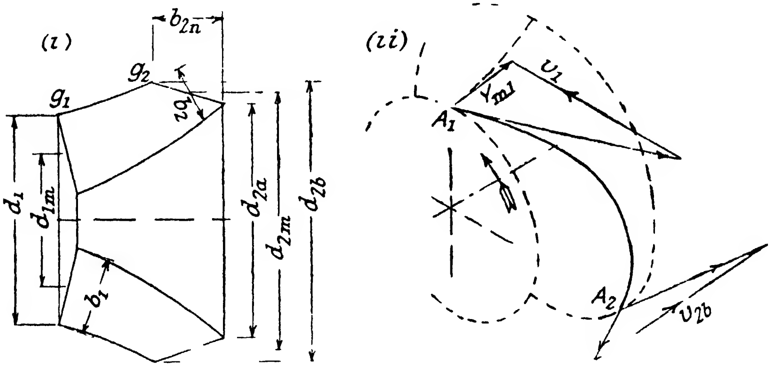


Fig. 71. Elements of screw-type rotor design

diameter of the screw-type rotor, Fig. 71. The usual range of values is :—

Speed ratio $\phi = \dfrac{u_1}{\sqrt{2gH_e}} = 1{\cdot}0$ to $1{\cdot}7$.

Flow ratio $\psi = \dfrac{Y_{m1}}{\sqrt{2gH_e}} = \dfrac{Q\Big/\frac{\pi}{4}d_1^{\,2}}{\sqrt{2gH_e}} = 0{\cdot}3$ to $0{\cdot}6$.

If the nominal *width ratio* λ is referred to the mean outlet diameter, d_{2m}, as in the diagram, then its value may range between

$$\lambda = \frac{b_{2n}}{d_{2m}} = 0{\cdot}25 \text{ to } 0{\cdot}3.$$

The ratio $\dfrac{\text{mean outlet diameter}}{\text{inlet diameter}} = \dfrac{d_{2m}}{d_1}$ is likely to have a value rather greater than 1·0 for medium specific speed pumps, and rather less than 1·0 for high specific speed pumps. The ratio

$$\frac{\text{inner outlet diameter}}{\text{outer outlet diameter}} = \frac{d_{2a}}{d_{2b}} \text{ may range from 0·7 to 0·9.}$$

Number of Blades : Rotor : 3 to 5,

Recuperator : 6 to 8.

There should preferably be an *odd* number of blades *either* in the rotor *or* in the recuperator, but the two numbers should never be equal. **(Example 13)**

107. Rotor Blade Angles. Ideal flow through mixed-flow rotors was described in § 34; departures from these ideal conditions were foreshadowed in § 41; and now we have to make still further simplifying assumptions in order to adapt the basic theory to practical use in design offices. The most important of these is, that along any transverse line of equal energy such as *ee* in Fig. 21 (i) the meridional velocity is *uniform.* In other words, the velocity variations mentioned in § 33 are to be disregarded, and a nominal mean value used instead, based on the transverse width of the rotor passages. Thus in Fig. 71 (i) the nominal mean meridional velocity Y_{m1} at inlet has the value $\dfrac{Q}{\pi d_{1m} b_1}$, and the corresponding value Y_{m2} at outlet would have the value $\dfrac{Q}{\pi d_{2m} b_2}$.

But it would be quite unjustifiable to assume that peripheral velocities are likewise uniform, so there still remains the need for computing the blade angles relating to a *number* of directive surfaces, § 34. Here it must suffice to study by way of example one such surface only, and it may conveniently be the outermost one, viz. the one struck out by the generating line $g_1 g_2$ in Fig. 71. The inlet and outlet blade angles may now be computed by the methods of § 93, always remembering that the rules for estimating the hydraulic efficiency η_h, and the whirl components V_n and V_∞, must now be interpreted only in the broadest sense.

108. Rotor Blade Shape. In Figs. 56 and 57 it was permissible to plot in a single plane all the angles and vectors involved. Now it is not possible. The nature of the problem can be seen from Fig. 71 (ii), where the plotting has been done by isometric projection. In conformity with the procedure of § 32, the inlet velocity triangle must be set in a plane tangential to the directive surface at point g_1; similarly the plane of the outlet velocity triangle must touch the directive surface at point g_2. Although instructive, the resulting diagram is valueless for design purposes; it does not suggest any practical way of laying out the curve joining the two points A_1A_2—the curve, that is, that will represent one of the blade edges visible in Fig. 62. The difficulty again reminds us of the limited utility of the customary type of projection used in Fig. 60 (i).

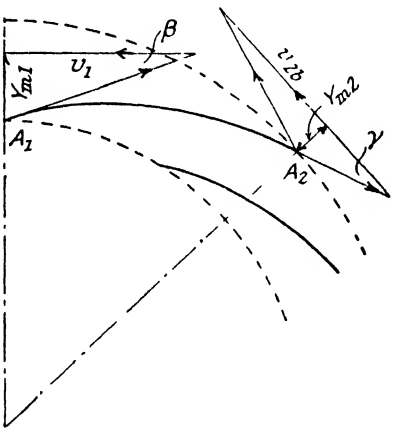


Fig. 72.—Velocity triangles for screw-type rotor.

A second simplifying assumption is thus necessary, viz. that the curve g_1g_2 in Fig. 71 (i) can be replaced by a straight line, from which it follows that the directive surface can now be taken to be *conical*, § 32. This conical surface can at once be developed by the ordinary rules of geometry, giving the figure plotted in Fig. 72. On this flat developed surface the velocity diagrams can be set off true to scale, and the blade outlines can be struck by circular arcs just as in § 95.

When the blade shapes for the other selected directive surfaces have been drawn by a similar process, the final setting-out of the blades in a form suitable for the pattern-shop might be done in the way suggested in Fig. 63, using a range of cross-sections as cut by a series of parallel planes normal to the rotor axis.

(*Notes* on the *recuperator* blades for half axial pumps are included in § 109.)

109. Design Data for Axial-flow Pumps. The nominal

values of the shape ratios to be used here can be based on the outer rotor diameter d_b and the inner rotor diameter d_a, Fig. 73.

Shape Number. Ranging in value from about 400 to 800, the shape number can be put in the form :—

$$n_s = 475\,\phi\sqrt{\psi}\sqrt{1-\left(\frac{d_a}{d_b}\right)^2}.$$

Gross Efficiency, η_m. Until recently few reliable figures for gross efficiency exceeded about $\eta_m = 0{\cdot}85$, and it would still be prudent to keep to this maximum when making preliminary estimates. In this connection it has to be remembered that test results for propeller pumps are peculiarly liable to error, §§ 160, 176.

Rotor Proportions. The range of shape ratios may be taken as :—

$$\text{Speed ratio} = \phi = \frac{v_b}{\sqrt{2gH_e}} = \frac{\pi d_b N/60}{\sqrt{2gH_e}} = 2{\cdot}0 \text{ to } 2{\cdot}7.$$

$$\text{Flow ratio} = \psi = \frac{V_a}{\sqrt{2gH_e}} = \frac{Q}{(\pi/4)(d_b^2-d_a^2)\sqrt{2gH_e}} = 0{\cdot}25 \text{ to } 0{\cdot}6.$$

$$\text{Diameter ratio} = \frac{d_a}{d_b} = 0{\cdot}5 \text{ to } 0{\cdot}6.$$

Blade Angles. Since the directive surfaces may now be assumed to be truly cylindrical, they may be developed ready for plotting the velocity diagrams as in § 36. The important correction to be applied to the diagrams of Fig. 23 (i) is the very uncertain one that links the true whirl component V_n with the ideal whirl component, V_∞.

Blade Number. The axial-flow rotor may have from three to five blades.

Blade Shape. The cross-sections in Fig. 73 give an impression of the change in inclination of the blade surfaces as the radius changes.

Recuperator. The number of recuperator blades is always *greater* than the number of rotor blades, and one or other of these numbers should be an odd one. Figs. 32 and 73 suggest how the inlet angle of the guide blades corresponding to each directive surface must accommodate itself to the absolute outlet velocity vector of the outlet velocity diagram. Manifestly this angle must relate to the *true* outlet whirl component, not to the

ideal one. Thus if δ is the required inclination, Fig. 73, then

$$\tan \delta = \frac{Y_a}{V_a}.$$

In a general way, these comments apply also to the co-axial recuperators of half-axial pumps, Fig. 67. **(Example 14)**

110. Aerofoil Theory of Propeller Pump Design. (*) The theories developed in an elementary fashion in §§ 38 and 39 may be of considerable practical utility to the designer. An alternative presentation of the elements concerned, taking into account the correction mentioned in § 41 (ii), is offered in Fig. 74. Diagrams (i) and (ii) are intended to show how the performance of an aerofoil may be affected when it is no longer solitary (i), but forms instead one of a group – a *grid* or *cascade*—of identical aerofoils (ii), all set at the same inclination α. The differences revealed by experiment are :—

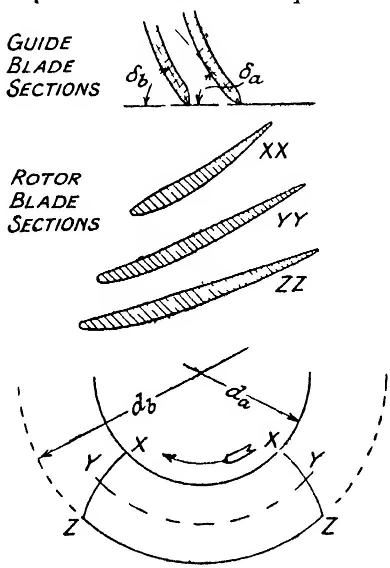


Fig. 73 — Blade cross sections for rotor and recuperator of axial flow pump

(1) The single blade has no ultimate influence on the direction of the liquid stream flowing past it. At a sufficiently great distance behind the blade, the velocity U of the liquid is in no way different from what it was originally, (i). The cascade of aerofoils (ii), on the other hand, does permanently deflect the entire liquid stream through an angle θ.

(2) Whereas the angle of attack α, and therefore the lift and drag, of a single aerofoil are related wholly to the magnitude and direction of the approach velocity U, the corresponding values for the cascade are more nearly dependent upon the velocity U_c which is the vectorial mean of the approach velocity U and the leaving velocity U_l. Although in Fig. 74, therefore,

the blades at (i) and (ii) have exactly the same inclination to an arbitrary datum line, yet the *effective* angle of attack has been reduced from α to α_c.

(3) The lift and drag on the aerofoils in cascade must now be regarded as the components of the total dynamic thrust, normal and parallel to the *effective or mean velocity* U_c. Almost certainly the values of the lift and drag coefficients will thereby be sensibly modified, although if revised values are not available

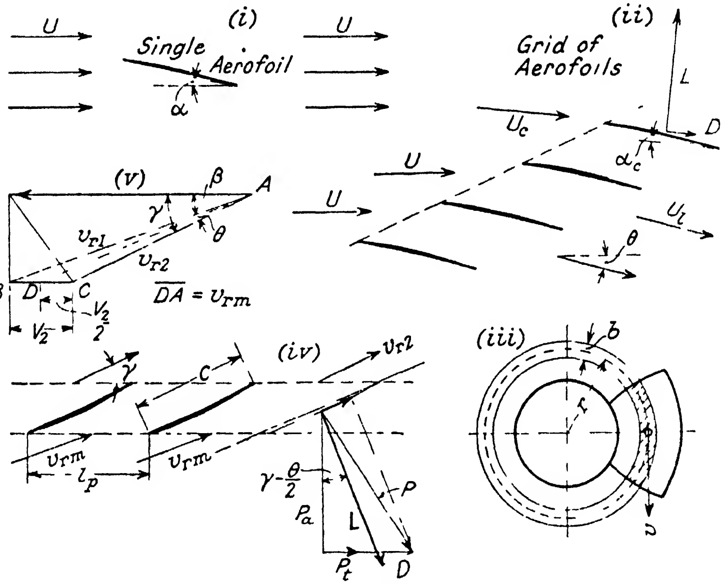


FIG. 71. Elements of aerofoil theory applied to axial-flow rotor blades.

the original ones may provisionally be accepted. Naturally these revised values will be based on the new or effective angle of attack, α_c.

111. Aerofoil Theory (*continued*). Armed with a knowledge of the characteristics of the grid of aerofoils, we can turn the grid over on to its back just as we inverted the single blade, Fig. 24. The resulting diagram, Fig. 74 (iv), now serves to represent also a developed sectional view of a set of pump blades comparable with Fig. 23 ; but here the blades are to be regarded as narrow elements such as the one shown by hatching in the plan view, Fig. 74 (iii). The corresponding combined velocity diagrams reproduced in Fig. 74 (v), relating to the

selected mean radius r, are the counterparts of the two sets of diagrams plotted in Fig. 23. Comparing the stationary cascade of aerofoils, Fig. 74 (ii), with the moving set of blade elements, Fig. 74 (iv), we observe that *absolute* velocities in the one correspond to *relative* velocities in the other thus :—

$$U = v_{r1}\,;\;\; U_l = v_{r2}\,;\;\; U_c = v_{rm}.$$

A sufficiently accurate method of estimating the mean relative velocity v_{rm} is suggested in diagram (v) ; we set off $BD = \frac{1}{2}BC$, and accept the vector DA as equivalent to v_{rm}. Further, the angle θ through which the stationary grid deflected the liquid stream is now represented in diagram (v) by the angle $\gamma - \beta$.

It is at this point that we begin to take a different view of the rotor blades and their purpose. Hitherto it has been something like this : we begin with a thin sheet of metal which we curve and bend until its edges just fit the angles of the velocity diagrams. Then we bend the sheet rather more to compensate for the unwillingness of the liquid to follow the curvature of the blade, i.e., we try to allow for the difference between V_n and V_∞. Finally we thicken up the blade a little to give it mechanical strength, make still another correction, and accept the resulting article as fit to take its place in the rotor. Now there is to be no question of flexibility ; right from the start we have to realise that we are dealing with a solid, unyielding piece of metal, whose shape will conform more or less to the cross-sections drawn in Fig. 73. Nevertheless there are three methods of controlling the behaviour of the blade :—

(i) We can choose whatever *shape* we like from the great variety of standardised sections developed by the aeronautical industry. For each shape there is information available to show how the lift and drag coefficients vary with the angle of attack—information which may be presented in graphs such as those in Fig. 75.

(ii) We can vary the *size* of the blade ; that is, still keeping the specified geometrical proportions of the selected cross-section, we can make the chord length c longer or shorter, Fig. 74 (iv).

(iii) We can vary the *inclination* of the blade, by tilting it about a transverse axis as in Figs. 69 and 70.

By making use of these various possibilities of adjustment, it is the purpose of the designer to ensure that :—

(*a*) At the chosen mean radius r the blade element will impart to the liquid the stipulated true whirl component V_2.

(*b*) The element will be big enough to impart energy to the liquid at the specified rate.

The one requirement is concerned with the *head* the pump develops ; the other with the *power* to be transferred. It is to be noted that the V_n . . . V_∞ problem has been completely dropped.

112. Application of Aerofoil Theory. The successive steps in an actual design computation are :—

(I) By the methods previously explained, estimate the rotor dimensions, the axial flow velocity Y_a, and the hydraulic efficiency η_h, § 109.

(II) Choose suitable values for the mean radius r and width b of a blade element, and compute corresponding values of the peripheral blade velocity v, and whirl component $V_2 = \dfrac{gH_e}{\eta_h v}$.

(III) From the plotted velocity diagrams, find the value of the angle of deflection $\theta = \gamma - \beta$, and the values of the magnitude and direction of the mean relative velocity vector, v_{rm}, Fig. 74 (v).

(IV) Having selected a likely aerofoil shape, use its characteristics when in grid formation to find the angle of attack α_c, relative to the mean vector U_c, Fig. 74 (ii), which will enable the grid to deflect the liquid stream through the stipulated angle θ. In the grid the pitch of the elements must have the correct value

$$l_p = \frac{2\pi r}{n}.$$

(V) The *inclination* of the blade element is now known : the element must be set at an angle α_c in relation to the mean relative velocity vector $v_{rm} = DA$, Fig. 74 (v).

(VI) From the aerofoil characteristic curves, Fig. 75, find the value of the life coefficient C_L and drag coefficient C_D *referred to the mean vector U_c and angle of attack α_c*. Since the lift L and the drag D on the element are proportional respectively to C_L and C_D, the vectors L and D can be plotted to

some arbitrary scale, as in Fig. 74 (iv). (Note that L is perpendicular to the mean relative velocity vector v_{rm}, while D is parallel to v_{rm}). The diagram is completed by adding the vectors P_a and P_t, perpendicular and parallel respectively to the direction of the blade velocity, v.

(VII) Evaluate *chord length* c in the following manner :—

Let E_b represent the *energy per second* imparted to the liquid by each of the n blade elements. Its value can be expressed in two different ways, thus :—

(i) Weight of liquid per second that each blade element acts upon

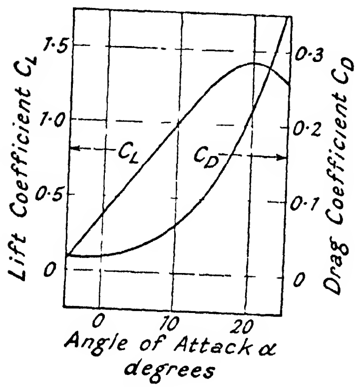


FIG. 75.—Typical aerofoil performance.

$$= w \cdot \frac{2\pi r b}{n} \cdot Y_a.$$

Energy per unit weight imparted to liquid

$$= \frac{V_2 v}{g} \quad . \quad \text{(from (II), above)}$$

Therefore

$$E_b = \left(w \cdot \frac{2\pi r b}{n} \cdot Y_a\right) \cdot \frac{V_2 v}{g} \qquad \textbf{(8-1)}$$

(ii) Since each blade element, moving with velocity v, encounters a resistance (*in the direction* of v) denoted by P_t, the work done per second in overcoming this resistance $= P_t \cdot v$.

That is, $\quad E_b = P_t v.$

Now the term P_t, viz., the tangential component of the total resultant dynamic thrust P on the element, Fig. 74 (iv), can be written

$$P_t = \left(\frac{P_t}{L}\right) \cdot L.$$

The ratio $\left(\frac{P_t}{L}\right)$ can be scaled off directly from the vector diagram.

The lift L has the value $C_L \cdot w \cdot b \cdot c \cdot \left(\frac{v_{rm}^2}{2g}\right)$ . . . (from equation 3-1).

Whence

$$E_b = \left[\left(\frac{P_t}{L}\right) \cdot \left(C_L \cdot w \cdot b \cdot c \cdot \frac{v_{rm}^2}{2g}\right)\right] \cdot v \quad . \quad \textbf{(8-2)}$$

Equating now the two values of E_b from equations (8-1) and (8-2), we find :—

$$\left(w \cdot \frac{2\pi r b}{n} \cdot Y_a\right) \cdot \frac{V_2 v}{g} = \left(\frac{P_t}{L}\right) \cdot \left(C_L \cdot w \cdot b \cdot c \cdot \frac{v_{rm}^2}{2g}\right) \cdot v.$$

In this identity, all the terms are known except C, the *chord length*, whose value can thus finally be extracted.

(VIII) Repeat the whole process for another blade element at a new radius r, and continue until the whole distance from inner to outer rotor radius has been explored.

113. Comments on the Aerofoil System. (i) It is an essential advantage of the procedure just outlined that for each of the various blade elements, each at a different mean radius r, a different type of aerofoil section may be chosen. Near the root of the blade the section might be relatively thick and stumpy, while near the rim it should preferably be slender.

(ii) A study of the force diagram, Fig. 74 (iv), makes it clear that the most favourable blade element shape is the one which has the least value of the ratio D/L. Although the expression $(P_t v - D v_{rm})/P_t v$ serves as a measure of the efficiency of the blade element, just as a similar expression has already done, § 39, yet we have here to note that this efficiency is necessarily higher than the hydraulic efficiency of the pump as a whole. The reason is this, that the energy input per unit weight $= \frac{V_2 v}{g} = H_e/\eta_h$ has to take into account not only the energy loss associated with the drag D, but *also* the energy loss in the recuperator, § 201. Nevertheless a knowledge of the blade efficiency gives useful guidance in forecasting the pump hydraulic efficiency.

(iii) From the force diagram we can also scale off the axial thrust P_a per blade element, and thus we can estimate pretty

closely the total axial thrust, ΣP_a, to be taken by the thrust bearing—a very important figure.

(iv) The necessary information relating to aerofoil performance *when a particular grid formation is specified* is in fact by no means easy to obtain ; and in any event the whole procedure calls for a good deal of experience and judgment in its actual application.

(v) At the end of a chapter largely devoted to the design and construction of axial-flow pumps, it may be worth while to turn back to the original diagram, Fig. 2 (iv), which showed the basic principle of generating pressure-head by forcing a row of blades transversely through a stream of liquid.

CHAPTER IX

PUMPS WITH MULTIPLE ROTORS

114. Why Multiple Rotors? According to the standard routine recommended in § 64, the first step in pump design is to find from the appropriate formula—or by any other likely method—the specific speed or shape number of the rotor. Hitherto we have conveniently assumed that the resulting number has fallen within permissible limits. But suppose that it does not? Suppose, for instance, that the stipulated pump duty could only be fulfilled by a rotor having a shape number of 20? This number immediately suggests a very narrow radial-flow (centrifugal) impeller, which, even if it could be made at all, would certainly have a very low efficiency for the reasons explained in § 197. At the opposite extreme it might happen that the only suitable rotor would appear to have a shape number of 1500, which again is far outside the range that experience has shown to be desirable.

It does not take long to resolve the problem if we take a broader view of it. Thinking in general terms of sources of pressure and sources of current, it becomes clear that we have only to group these units in series or in parallel in order to generate any desired pressure or flow. Thus a group of three electric secondary cells in series, Fig. 76 (i), will generate the same current as a single cell, but three times the pressure-difference. A parallel arrangement, Fig. 76 (ii), yields the voltage of one cell but the current of three. In rotodynamic terms the unit will be one rotor with its diffuser; it may be represented pictorially by a rectangular solid or block, whose three dimensions roughly correspond to head, discharge, and speed. A

low-specific speed pump would be suggested by a tall, thin block, while a broad, squat block might indicate a high-specific speed pump. If a single block does not generate all the pressure we want, then we pile blocks one above the other, Fig. 76 (iii) ; if the basic unit fails to yield the stipulated flow, then doubtless a group of units set side by side will do so (iv).

115. Multiple Pumps or Multiple Rotors ? By looking at the specific speed formula $N_s = K_s \dfrac{N\sqrt{Q}}{H^{\frac{3}{4}}}$, it becomes clear that rotors in *series* will be needed if the original provisional value of the specific speed was too *low*, while the parallel arrangement will correspond to too high a value. The *number* of rotors, m, can be found thus : –

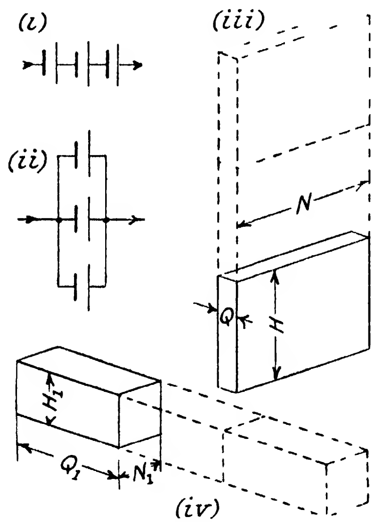


FIG. 76. – Schematic representation of units arranged in series and in parallel.

For the series arrangement, we write

$$N_{ss} = K_s \cdot \frac{N\sqrt{Q}}{(H/m)^{\frac{3}{4}}}$$

where N_{ss} represents the lowest permissible specific speed per rotor.

For the parallel arrangement, we write

$$N_{sp} = K_s \cdot \frac{N\sqrt{\dfrac{Q}{m}}}{H^{\frac{3}{4}}}$$

where N_{sp} represents the highest permissible limit. After extracting the value of m, the nearest suitable whole number is accepted.

The electrical analogy continues to be useful when we approach the practical problem of conveniently arranging the groups of rotors. The three cells shown in Fig. 76 (i) may either be independent units linked by external wiring, or we may put the three in a box with internal connections and call the assembly a battery. Similarly, high liquid pressures may be built up either by coupling together separate pumps by external

lengths of pipe, or by collecting into a single casing the individual rotors. The self-contained type of assembly is termed a *multi-stage pump* to distinguish it from the *single-stage pumps* hitherto studied. There is no special name for a pump which has two rotors in parallel.

It is only by studying the general circumstances of the whole installation that one can decide whether a group of individual pumps or a single multiple-rotor pump would be most suitable. When large quantities of water are to be discharged against a low head, probably two or more standard single-stage pumps arranged across a pair of inlet and outlet " bus pipes " will best meet the conditions. But for generating high pressures the multi-stage pump is the more usual expedient. In any event consistency requires that in this book pumps with multiple rotors shall alone be described in the present chapter, while dispositions of pumps in series or parallel are studied in Part D, §§ 318-322.

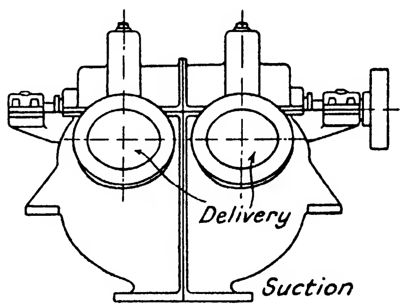


FIG. 77. Low-lift centrifugal pump with twin impellers.

116. Pumps with Rotors in Parallel. As has just been implied, this arrangement is not often used for ordinary services. The reason lies here : that for a stated head, speed, and discharge, a screw pump or an axial-flow pump would probably be more compact and less expensive than a twin-impeller centrifugal pump. But this alternative may have to be rejected because the single-rotor high-specific speed pump has the wrong kind of characteristic, § 213, or would refuse to draw against the stipulated suction lift, § 253, or would fit in awkwardly with the piping layout. Fig. 77 gives an impression of a twin-impeller pump. Water enters through a single large suction branch and then divides itself between the two rotors, each of which is of the double-entry medium specific-speed type. Each of the volutes has its own outlet flange, and no doubt to these flanges an outlet *Y*-piece or breeches-pipe would be bolted so as to marshal the water into a single stream again. Sometimes

the pump casing may have individual suction branches for the two impellers, with an inlet Y-piece.

The most likely field for twin-impeller pumps is in graving-dock installations, circulating-water systems in steam power stations, and so on, where the discharge per pump is of the order of 10,000 to 20,000 g.p.m. or more. Pumps each with three impellers in parallel have been built in the past, but it is extremely unlikely nowadays that they would offer the best solution to any pumping problem.

A special application of the twin-impeller principle is seen in some kinds of condensate-extraction pump, Fig. 97 (i).

Multi-stage Pumps

117. Problems of Design. When the rotors are arranged in series: when the liquid flows through each of the rotors in turn: then each rotor with its accompanying recuperating element is termed a stage. Thus a six-stage pump has six rotors mounted on a common shaft, and six recuperators. One of the main difficulties in design arises when planning the passages that lead the liquid from the outlet of one stage to the inlet of the next stage. Hydraulic and mechanical considerations are here directly at variance. In order to make the pump stiff and compact, we should like to pitch the impellers as close together on the shaft as possible. But will that not entail serious hydraulic losses? In a normal single-stage pump it is essential to treat the liquid with the utmost solicitude from the moment it leaves the rotor to the moment when it is safely down the delivery pipe. During the whole of this period we are trying to coax from the liquid more and more pressure energy, in exchange for the excess velocity energy the rotor has given it. If these carefully-disposed outlet passages or recuperators are now bent brusquely inwards in order to divert the liquid back again towards the impeller inlet, we can be sure that the liquid will signify its displeasure in the usual manner. It will throw away energy. In consequence we find that even the most skilfully-designed multi-stage pump almost invariably has a lower gross efficiency than a comparable single-stage pump having the same *shape of rotor*. But of course the multi-stage pump is much *more* efficient than a

single-impeller pump having the same discharge and shaft speed, and generating the same head.

In other respects the multi-stage pump is a good deal more compliant than the single-stage machine. It is a great convenience to the designer to be able to control not only the shape and size of the rotors, but their number as well. Then he has also a greater variety of ways of resisting or neutralising axial thrust, §§ 76, 77. In addition to the methods there described, there is scope for automatic hydraulic devices.

118. Range of Types. (i) *Rotors.* Impellers of the low specific speed, side-inlet centrifugal type, § 69, are almost invariably chosen. Double-inlet impellers are occasionally used but will not again be mentioned in this book. A combination of a double-inlet impeller with two side-inlet impellers is sometimes preferred, e.g. in very large hydraulic-storage pumps, § 157.

(ii) *Casing.* The complete pump assembly may be of the

(*a*) Split-casing type, or

(*b*) Barrel type, or

(*c*) Ring type.

In the first of these the casing is divided in an axial plane, while in the other types the joints are transverse to the axis.

(iii) *Recuperator.* The volute type, § 45, is more suitable for split-casing pumps, while the diffuser or guide-vane type, § 44, is acceptable for ring or barrel pumps.

(iv) *Disposition of impellers.* Opposed impellers (back-to-back) are generally found in split casings ; ring and barrel casings usually contain uni-directional impellers, all having the same aspect.

(v) *Horizontal or Vertical Shaft?* The standard pumps described in this chapter almost invariably have horizontal shafts, but the special multi-stage pumps designed for working down boreholes naturally have vertical shafts, § 130.

119. Split-casing Two-stage Pumps. Opposed impellers working within horizontally-split volutes offer by far the most convenient and most popular arrangement of two-stage pump. Here in Fig. 78 we see the two identical rotors each in its own volute, and we next have to make the difficult decision foreshadowed in § 117, viz., how to conduct the liquid from the outlet

of the first stage to the inlet or eye of the second stage impeller. Two general solutions are illustrated in Fig. 79.

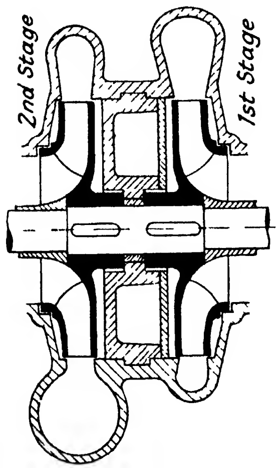


FIG. 78.—Arrangement of opposed impellers in two-stage pump.

(i) An external and quite separate U-shaped transfer pipe couples the first stage delivery flange to the second stage inlet flange. The descending coned limb of the transfer pipe permits the final operation of energy recuperation to be effectively carried out, and thus the second duty of the pipe, to direct the liquid back to the pump again, is only begun when the liquid has no more available velocity energy left (diagram (i)). We should therefore expect a relatively high efficiency from such a pump, an advantage which is naturally not a free gift: the pump clearly calls for a special bed-plate and foundation-block.

(ii) When the transfer passage takes the form of a port cast in the upper or lower half of the casing, then we arrive at a standard commercial product, Fig. 79 (ii).

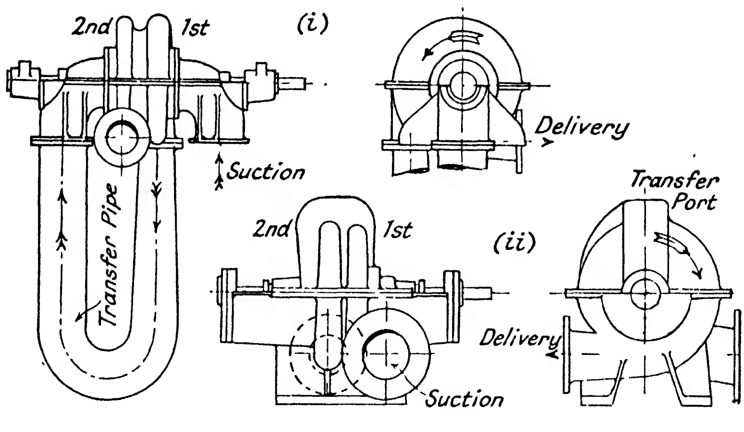


FIG. 79.—Types of split-casing two-stage pump.

Otherwise the pump almost exactly resembles a normal split-casing single-stage machine, § 71. Such two-stage pumps are available in sizes up to say 5 to 6 in., for total heads up

to several hundred feet. (Used in this sense, the dimensions relate to the diameter of the pump branches.)

120. Other Split-casing Pumps. The split-casing disposition has two such important points in its favour that it is applied also to pumps with as many as eight stages. As is already evident, these points are : (i) very great reduction of axial hydraulic thrust, which in an ideal pump would amount to total elimination, (ii) simplicity of construction and ease of access in use. The whole inside of the pump is laid open for inspection by just lifting off the top half of the casing. In the 6-stage pump illustrated in Fig. 80, the inter-stage communications are formed of a combination of the two systems described in § 119 above. There are internally-cast transfer ports between

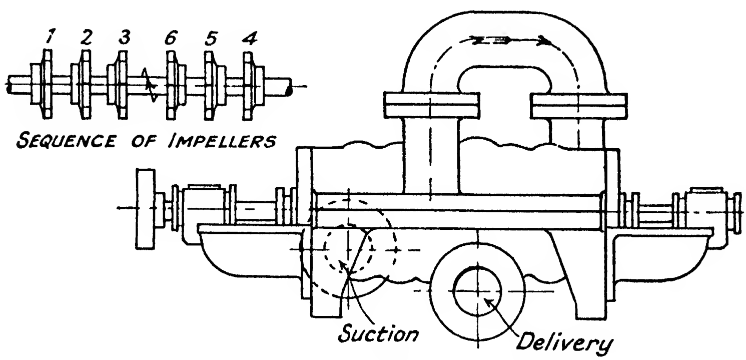


FIG. 80 Six stage pump, split casing type.

stages 1 to 2, 2 to 3, 4 to 5, and 5 to 6, while an external transfer pipe leads the liquid from stage 3 to stage 4. But now we have to examine possibilities of leakage which, although present in opposed-impeller 2-stage pumps, might there only have been relatively trifling. Referring to Fig. 78, the liquid will evidently try to leak along the shaft from the back of the second stage impeller to the back of the first stage impeller, and only by maintaining a sufficiently fine clearance between shaft and inter-stage diaphragm plate can the leakage be kept within reasonable limits (see § 83) (**Example 16**). Referring also to Fig. 79, we observe that the gland of the second stage of each pump must necessarily be exposed to a considerable positive pressure.

When these tendencies are magnified as they are in the 6-stage pump, Fig. 80, it may even be worth while altering the

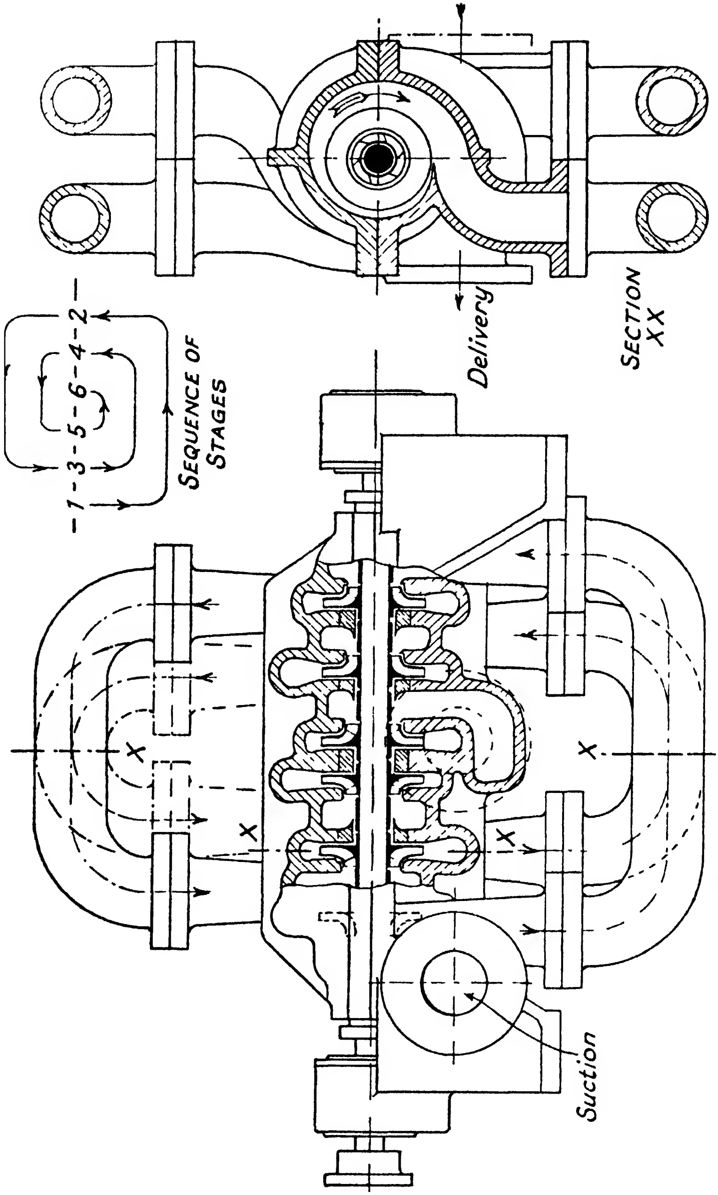


FIG. 81.—Improved type of split-casing multi-stage pump.

whole design to try to minimise them (*). The resulting construction is shown in Fig. 81. External transfer pipes are used for all inter-stage communication except between the last two stages, 5 and 6. Whereas in the original design there was a pressure-difference between stages 3 and 6 amounting to one-half of the total pressure on the pump, in the modified design the pressure-difference between adjacent stages is nowhere more than that due to a single stage. Similarly the right-hand gland and stuffing-box in Fig. 80 must sustain in equivalent conditions three times the pressure that it does in Fig. 81; and problems of maintenance may thus ensue such as mentioned in § 141. A final point to be noticed in Fig. 81 is the opposed disposition of the volutes in a *radial* direction. In no volute pump is it possible to guarantee that the impeller will not have to sustain a transverse thrust, §§ 73, 207, and it is all the more important to be on the watch when the thrust may fall upon a long shaft supported by two widely-spaced bearings, as in multi-stage pumps. The disposition in Fig. 81 is such that for each pair of adjacent impellers, the radial thrusts are likely to neutralise one another.

If it be suggested that the design looks bulky and complicated, we have to remember that energy inputs exceeding 100 or 200 h.p. are now in question, and that even ½ per cent. improvement in efficiency might justify a very sensible increase in constructional cost.

121. Barrel Pumps: Ring Pumps. As its name implies, the casing of the *barrel* pump has no longitudinal joint, but is virtually a cylinder with flanged ends to which the bearing housings are spigoted, Fig. 82. The impellers all point towards the suction end of the pump, and the intercommunication passages are all formed within disc-shaped castings which fit snugly inside the barrel. Liquid from any one impeller first flows through the diffuser or recuperator part of the passages, § 44; then after having had its direction reversed, the liquid returns inwards towards the eye of the next impeller. As the liquid is drawn off uniformly all round the rotor periphery, there is little likelihood of transverse thrusts developing such as we find in volute pumps; in regard to the hydraulic axial thrusts of the various impellers, these are now cumulative and are wholly absorbed by a special automatic device, § 123.

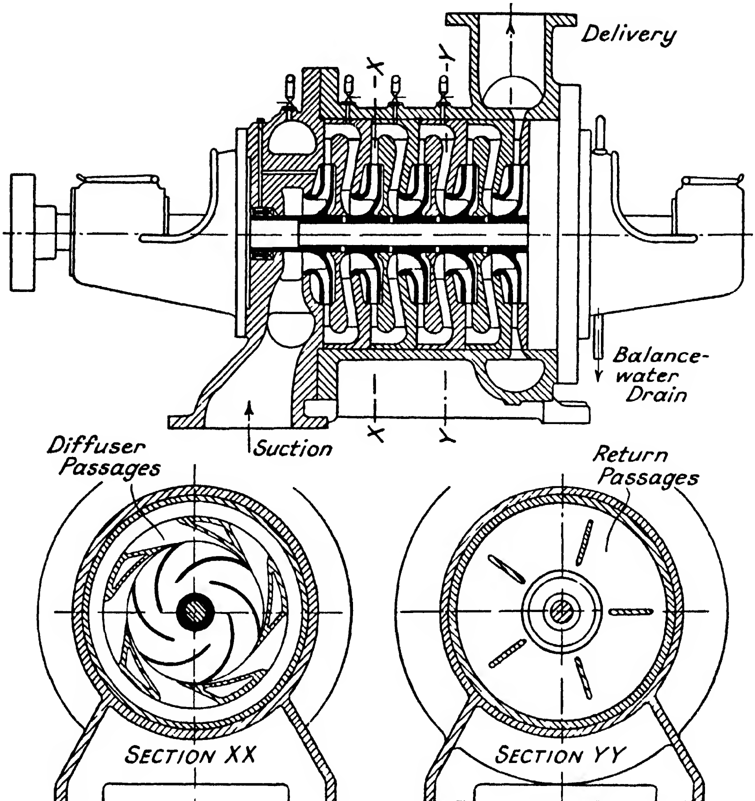


Fig. 82.—Barrel-type multi-stage pump.

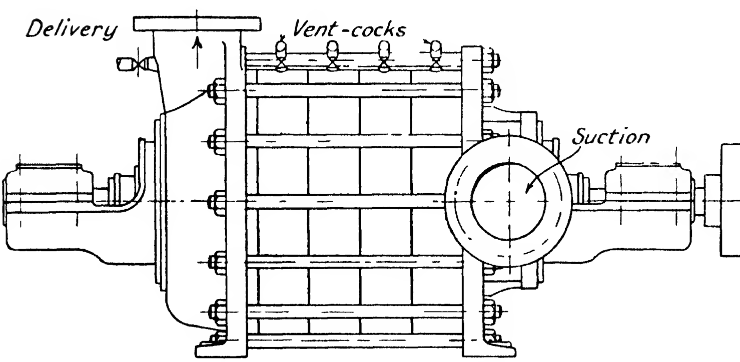


Fig. 83.—Ring-type multi-stage pump.

The *ring type* of pump differs from the barrel type only inasmuch as the cylindrical casing containing the diffuser discs is now discarded, the elements being clamped together only by external tie-bolts, Fig. 83. From a standard design of disc and standardised inlet and outlet castings, therefore, pumps suitable for a wide range of pressures may be built up by varying the number of discs or stages.

The purpose of the *vent-cocks* seen in Figs. 82, 83, is to release entrained air when starting the pump, §§ 291, 354.

122. Diffusers and Return Passages. Here we come to the detailed study of what was described in § 117 as one of the cardinal difficulties in multi-stage pump design (*). The mechanical problem has been neatly solved; as Figs. 82 and 83 make clear, the multiple impellers are very compactly and conveniently housed. Is the solution of the hydraulic part of the problem equally satisfactory? Functionally the complete diffuser disc has three elements: (i) the diaphragm plate which separates adjacent impellers, (ii) a set of diverging passages for the outward-flowing liquid, and (iii) a set of passages for the inward-flowing liquid. Constructionally we may form these elements in a single casting, or the diffuser disc may be of composite nature.

The single casting suggested in Fig. 82 is manifestly fit for small pumps only. As the thin and sharply-pointed guide blades must be made of bronze or similar metal, then that expensive metal must be used for the whole casting. The radial blades of the return passages also have rather a doubtful look. If the outlet diffuser blades have not succeeded in removing the whole of the whirl component from the liquid, there is a likelihood of eddy losses when the liquid impinges on the inlet edges of the return blades.

One form of composite construction is shown in Fig. 84. The diffuser itself is of such simple form that it may be machined or at least polished all over. The blades are so relatively thin that the whole circumferential space between them is hardly different from a whirlpool chamber, § 43 (ii). As for the main cast-iron disc into which the bronze diffuser ring is recessed, this also carries neck-bushes for the impeller eye and for the impeller boss; and the webs forming the return passages are now suitably curved instead of being radial. In passing from

the diffuser guides to the return passages, the liquid traverses a circumferential chamber which is free of all obstruction, and

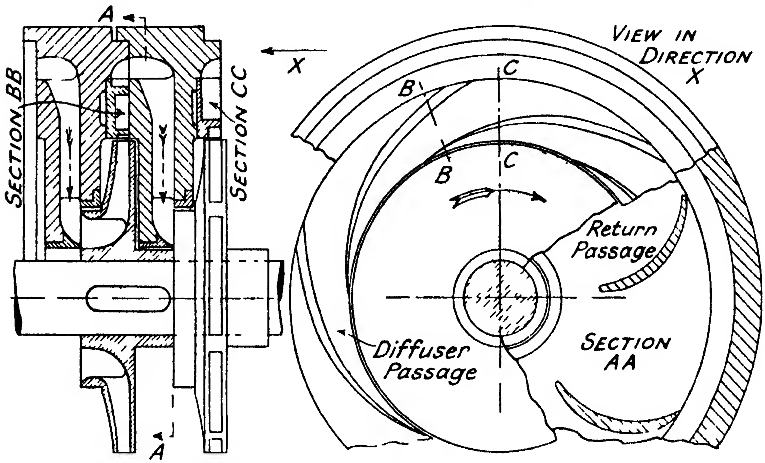


FIG 84 Details of diffuser ring and return passages

which thus gives the liquid the most favourable conditions for reversing its radial direction as well as for converting into pressure energy some of its residual velocity energy.

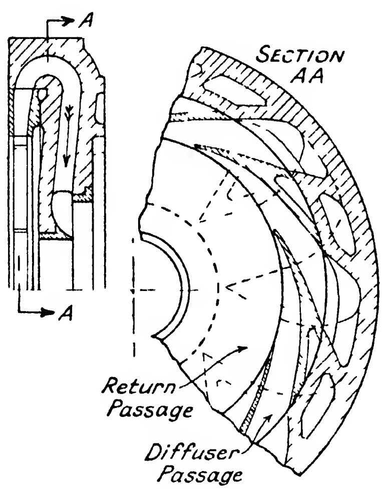


FIG 85. Alternative form of diffuser and return passages

An alternative theory governs the design of the composite diffuser disc sketched in Fig. 85. From the moment the liquid leaves one impeller almost to the moment at which it reaches the next, it is closely guided by the walls of a series of continuous passages; it is given no such opportunities of choosing its path as it had in Fig. 84. Another difference which is purely constructional is that in the diffuser casting the passages are now enclosed on both sides, thus making a smooth finish more difficult to attain. But in both cases, recuperation

is improved by " flaring " the cross-section, instead of keeping uniform axial width as in Figs. 26, 27.

Comparison between Figs. 85 and 84 indicates that in the former the mean absolute velocity of the liquid will be higher than in the latter; but this may be balanced by the reduced risk of eddy losses. The two examples of disc here illustrated represent only a selection from quite a wide variety. They are equally suitable for barrel or for ring pumps.

123. Axial Thrust. The back-to-back arrangement of pairs of impellers or groups of impellers, §§ 119, 120, must not be relied upon to neutralise axial thrust completely. There is always a chance that inequalities of construction, variations of shaft diameter (collars, shoulders, etc.), and the like, § 74, may impose an unbalanced load on the shaft of several hundred pounds. Since, moreover, the thrust may come from either direction, the bearing that resists it must be designed accordingly.

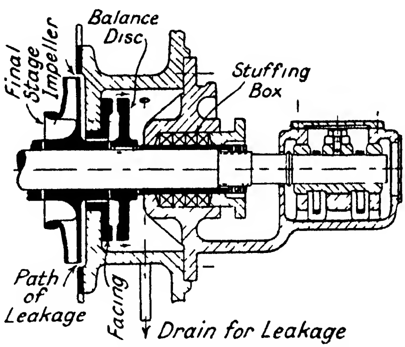


FIG. 86. Hydraulic balancing system

There is no such uncertainty when uni-directional impellers are used, Figs. 82, 83; the cumulative thrust will be steady and it will be considerable. If a ball thrust bearing can be found that will stand up to the combination of high speed and heavy load, then nothing better could be desired. But a hydraulic balancing device is usually preferred, because it is wholly automatic and requires no attention. Its principle of operation is illustrated in Fig. 86. Between the last-stage impeller and the stuffing-box at the delivery end of the pump there is formed a chamber which houses a *balance-disc* keyed on to the shaft; it works against a stationary facing of suitable metal fixed to the casing. Liquid can escape from the chamber either freely to the atmosphere, Fig. 82, or it may be taken back to the suction side of the pump. In consequence there is a pressure-difference, between the liquid behind the last impeller

and the liquid within the chamber, amounting roughly to the total head on the pump.

124. Theory of the Balance-disc. The diagrams in Fig. 87 show how the entire rotating assembly—shaft, impellers, and balance-disc—can adjust itself automatically to suit the gross thrust P_g on the impellers. Let us suppose first that the disc is hard up against its facing, so that no liquid can escape past its rim. That will mean that the pressure acting on the inner surface of the disc is equivalent to the delivery pressure-head H_{md} in the pump. Now the diameter of the disc is calculated so that this pressure will create a total resultant thrust acting *away* from the pump whose magnitude is *greater* than the impeller thrust P_g acting in the opposite direction. In consequence the disc will be forced slightly away from its seating, as in Fig. 87 (i). Liquid immediately begins to leak through the small running clearance between casing and impeller-boss, and then through the clearance between the disc and the casing, Fig. 86. As explained in §§ 82, 83, the resulting drop of pressure head h_{lc} will depend upon the rate of leakage; if the disc moves still further to the right, as in Fig. 87 (ii), leakage will increase, the pressure drop will increase to h'_{lc}, and in turn the pressure-head acting on the disc will *diminish* to h'_{ld}. Since the total thrust on the disc is proportional to wh_{ld}, it is easy to see that in practice the rotating assembly will float or adjust its longitudinal position until the rate of leakage creates just such a pressure drop as will correspond with a thrust on the disc exactly equal to the thrust P_g on the impellers. The liquid itself serves as a lubricant between balance-disc and casing: there can be no metallic contact here.

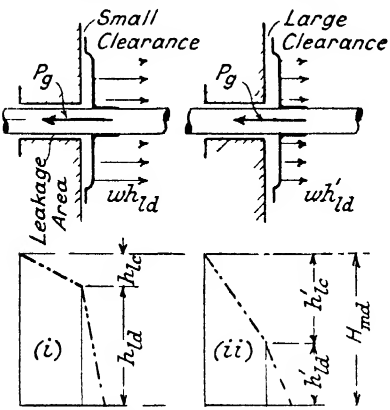


Fig. 87 Pressure changes in hydraulic balancing system.

The hydraulic balancing system has the further advantage of relieving the delivery stuffing-box of the high pressure to

which it would otherwise be subjected, Fig. 86; but on the other hand the leakage of liquid represents a quite appreciable waste of power, § 192. **(Example 17)**

125. Design Data for Multi-stage Pumps. Only those points are mentioned here in which the multi-stage pump differs from the single-stage centrifugal pumps described in §§ 90 to 95.

Number of Stages. The number of stages, m, as computed by the method explained in § 115 should preferably not exceed 8 or 10. The greater the number the longer the shaft must be and the bigger its diameter, resulting in increased inter-stage leakage and impaired efficiency. Constructional and operational difficulties also mount up. Instead of using an excessive number of stages in one pump, it may be preferable to use two separate pumps in series, § 320 (*b*). The maximum permissible *head per stage* may provisionally be taken as 500 ft.

Shape Number. It is important that in all specific speed and similar formulæ, the *head per stage* should now be inserted and *not* the total head on the pump. Thus if the total effective head is H_e, then the shape number *per impeller* will have the value

$$n_s = \frac{1000(N/60)\sqrt{Q}}{(gH_e/m)^{\frac{3}{4}}} \quad . \quad . \quad . \quad \S\ (59)$$

This value usually ranges between about 50 and 80.

Gross Efficiency. Comparing a single-stage pump with a multi-stage pump using identical impellers, the multi-stage pump will always have a lower gross efficiency, under comparable conditions, for the following reasons: (i) there is an inevitable energy loss in the transfer ports or passages, § 117, (ii) there may be a loss of energy to be debited to the balance disc, § 124, (iii) there will inevitably be inter-stage leakage, § 120. Here is a rough method of allowing for such losses: From the selected shape number *per stage*, and from the known water horse-power *per stage*, we estimate from Fig. 53 what the gross efficiency for an equivalent single-stage pump would be. If we then deduct from this value about 3 to 5 per cent., the result should serve reasonably well as a preliminary estimate. In brief,

$$\eta_m \text{ (multi-stage)} = \eta_m \text{ (single-stage)} - (0{\cdot}03 \text{ to } 0{\cdot}05).$$

Rotor Proportions. Fig. 54 will give the desired information, but it might be well to take a rather higher value for the diameter ratio $\frac{d_1}{d_2}$ to allow for the relatively greater diameter of the multi-stage shaft. This implies that the checking of the area of the impeller eye, § 94, is particularly necessary now.

Overall Proportions. In order to form a general impression of the size of the complete pump, one may use the following ratios :—

If d_3 is the outside diameter of the diffuser rings, Fig. 88,
d_2 is the impeller diameter,
l_p is the axial pitch of the impellers,
then d_3/d_2 varies from about 1·5 to 1·7,
and l_p/d_2 varies from about 0·3 to 0·4.

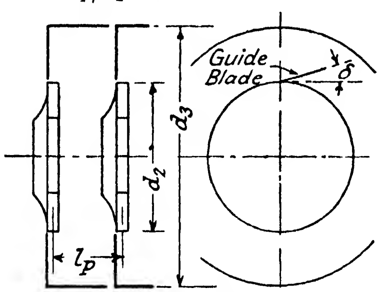


FIG. 88. Diagram illustrating proportions of multi stage pump

Diffuser Blade Angle. Although ideally the inlet tip of the diffuser blades, § 122, should be parallel with the true absolute velocity vector U_2 which represents the liquid leaving the impeller, yet it is found preferable to make the blade angle slightly greater than is thus indicated. The ideal value of this angle δ is $\tan^{-1}\left(\frac{Y_2}{V_\infty}\right)$; the actual angle should be about $1{\cdot}1 \tan^{-1}\left(\frac{Y_2}{V_n}\right)$.

Bearings. Two external sleeve bearings, § 78, are obligatory for any pump fitted with a balance-disc, e g , Figs. 82, 83 and 86. Locating-collars or the like must not hamper the axial play of the rotating element.

For opposed-impeller pumps, either sleeve journal bearings, Fig. 79 (i), or ball or roller bearings, Fig. 79 (ii), 80 and 81, are suitable. The bearings must be able to take up whatever residual end thrust may develop. **(Example 15)**

126. Construction and Assembly. Because of the widely-spaced bearings, and the numerous masses disposed along the shaft of a multi-stage pump, the *balancing* of the impellers must be done with special care, § 96. Not only must the individual

impellers be accurately balanced, but the balance of the entire rotating parts when finally assembled must also be checked.

On examining afresh such drawings as Figs. 80 to 83, it becomes clear why the split-casing type of construction is preferred in the workshop. There do not seem to be special difficulties in fitting the rotating element into place and finally closing the casing. At the other extreme is the barrel type of pump, which looks as though very experienced hands will be needed to put it together. The impellers and the diffuser discs must be threaded alternately on the shaft, and the discs in turn fitted into the barrel. If the casing is bored truly cylindrical, some of the discs will have to be forced a long way through the barrel before they come to their destined position. It may be still more troublesome to get them out again when the time comes to dismantle the pump for overhaul or repair. If, however, the successive discs are of slightly different diameters, and the barrel is bored to suit, each disc will pass easily along the barrel until it comes to its own seating.

A similar problem arises when mounting the impellers on the shaft, and a similar solution is available. In regard to the keys that secure the impellers, sometimes a single long key is provided, and sometimes each impeller has its own key.

127. Other Types of Multi-stage Pumps. It is not only centrifugal types of impeller that can be arranged in series to form multi-stage pumps; mixed flow and axial-flow rotors may be similarly disposed. The recuperators chosen for multi-stage *borehole pumps*, § 132, sometimes resemble those found in normal single-stage *half-axial* pumps, § 102. As many as six small *axial-flow* rotors may be mounted on the same shaft, or two large rotors. If the conditions are such that variable-pitch propellers are desirable, § 104, and if a single rotor will not develop the desired head, then the complete pump may have two such rotors mounted in series on one shaft.

CHAPTER X

PUMPS FOR SPECIAL DUTIES

128. What are Special Duties? Previous chapters have described the construction of pumps installed at ground level and intended generally for handling clean cold water or similar liquids. This present chapter will go on to suggest some of the modifications in design that may be required if these standard pumps are no longer suitable. It is reasonable to expect, for instance, that a normal pump might not be happy if it were set to work at the bottom of a shaft 2000 ft. below ground level; or again, the pump might protest if asked to handle boiling water, hot oil, concentrated acids, crude sewage, sandy or gritty water, or the like. Special-purpose pumps, then, might be grouped into four general categories, thus:—

(i) Underground pumps,
(ii) Pumps for hot liquids, etc.,
(iii) Pumps for impure or corrosive liquids, etc.,
(iv) Miscellaneous pumps, e.g. self-priming pumps, hydraulic-storage pumps, etc., etc.

It may be found that sometimes quite trivial alterations to a standard design will suffice, while to meet more severe conditions a completely new style of pump will be needed.

Underground Pumps

129. Well-pumps : Mining Pumps, etc. A rotodynamic pump established at ground level can only lift water from a well if the total suction head is kept within sharply-defined limits, § 253. This means in effect that the water level cannot be more than about 15 or 20 ft. below ground level. For all greater depths, the pump must be taken below ground so as to be within reach of the water. This is not difficult to do if small quantities of water are to be drawn from a dug well ; a standard horizontal pump can be lowered down the well on to a platform just clear of the water surface, the installation being in every way normal. Trouble is only likely to arise if the water level fluctuates excessively, either due to seasonal variations or because pumping itself draws down the level. Then it may happen that if the pump is set low enough to reach the water under the worst conditions, it may be drowned when pumping ceases and the water level rises again (see also Fig. 207, § 312).

Mining Pumps have the duty of forcing to ground level the water that percolates into mine workings and that would otherwise flood them. If there is no existing gallery close to the drainage sump which could accommodate the pumping set, then a special one can be driven. An electric multi-stage horizontal pump will usually be suitable, §§ 117 to 121, with working parts of special metal to resist the action of the slightly corrosive water ; e.g. the impellers may be of acid-resisting bronze, monel metal, etc., and the shaft may be made of or protected by similar material. If two multi-stage pumps are set in series, the high-pressure conditions imposed on the gland of the second pump may have to be resisted by the methods described in § 141. To protect the balance-discs from damage by sand or mud in the pumped water, they may require an independent supply of filtered water

Sinking Pumps are used for sinking a mine shaft or a well through water-bearing strata, or for unwatering old workings, foundations, and the like (*). It is thus essential that the pump should be able to follow the water downwards as the work proceeds : it can finally be withdrawn when the permanent pumps are installed, or sometimes the sinking pump itself is left in place to continue pumping. As a rule the complete outfit

consists of a vertical-spindle electrically-driven pumping unit, mounted in a frame built of rolled sections. The whole is slung from a derrick or head-work above the shaft. Flexible leads take supply-current down to the motor; and as the set is progressively lowered, successive lengths of piping are added to the rising main which delivers the water to ground level. Almost always a multi-stage pump will be needed, and the internal parts will have to deal with water that may be gritty as well as possibly slightly corrosive. But basically a standard design of pump will serve: it is rather the motor that will require special protection to fit it for very rough outdoor conditions.

130. Borehole Pumps. When water has to be raised from boreholes instead of from open wells, there can no longer be any question of adapting standard pumps: the designer is now faced with a completely new set of problems (*) If we include also the tasks of the civil engineer, there are three problems: (i) how to get the water from the subsoil into the borehole, (ii) how to make a pump small enough to pass down the borehole, (iii) how to transmit energy down to the pump when it is finally installed below ground. All these questions ultimately depend upon the general relationship between the *diameter* of the borehole and the *yield* from it. In ideal conditions, when the borehole is drilled through uniform permeable rock, a mathematical connection can be established between diameter and yield: the yield increases more slowly than the diameter, i.e., doubling the diameter does *not* double the yield. Admittedly this interconnection has little relation to actual fact, which must take into account the headings, galleries and adits that may be driven to supplement the yield by tapping copiously-flowing fissures and so on; yet nevertheless the results from actual installations show that in comparable conditions the total yield from a borehole is unlikely to increase as fast as the *square* of the diameter.

Now consider the correlation between diameter and discharge of rotodynamic pumps of a given *shape number* running at a given shaft *speed*. From § 56, we find that the output varies as the *cube* of the diameter. The flow *into* the borehole, then, may vary at most as the square of the diameter; the flow *out of* the borehole can be assumed to vary as the *cube* of

the diameter. It clearly follows that the pump designer's troubles are only likely to be acute with small bores, say of 12-in. diameter or less ; here the water may flow into the hole quicker than he can get it out. But when the estimated demand for water implies a borehole of 30-in. diameter or more, there should no longer be any need to skimp the pump external diameter ; if this is proportioned according to ordinary rules, there will be little fear that the pump will be too big to enter the well.

131. Methods of Drive. There are two possibilities here : (i) the driving agent or apparatus may be set above ground, transmitting energy down to the pump by means of a rotating shaft ; or (ii) the driving apparatus may itself go down the borehole to join the pump, the two forming a close-coupled unit resembling a sort of sinking-pump, § 129. Indeed such units sometimes are used as sinking-pumps. One of these possibilities is at once ruled out if an oil-engine or steam-turbine is the source of energy—the engine must stay on top. The choice then remains between above-ground or below-ground electric motors. And it must be remembered that below-ground may often mean below water, for the reasons given in § 129, which now may be far more potent. The total annual range in water level in the borehole may amount to 100 ft. or more, and therefore in the case of the close-coupled underground electric motor there is the probability that the motor as well as the pump will be drowned.

Which, then, is the better : a motor above ground, comfortably housed and able to receive every attention and consideration that is due to it ; or a motor deep below ground, out of sight, out of reach, and continuously working under just those aqueous conditions that revolt every instinct of the mechanical engineer ? As already suggested, the size of the bore is again the controlling factor, but tendencies are directly opposite to those that affected the pump itself. It is the large boreholes that may worry the motor designer. It is when the power output reaches several hundred horse-power—just when the pump designer begins to breathe freely--that the motor designer feels cramped. Probably even if he could produce a motor that would fit inside the borehole, it would overheat in service. On the other hand, when the energy required by a

small-bore pump is hardly more than a few horse-power, it could be delivered without undue difficulty by a motor of the requisite small diameter.

So one may say *in very general terms* that if the pump absorbs more than about 100 h.p., it should be driven by shafting from an electric motor installed at ground level, while a pump of less than 50 h.p. may preferably be driven by a direct-coupled motor stationed down the well. The two complementary—and sometimes competing—types are known respectively as :—

(i) Shaft-driven borehole pumps, and

(ii) Submersible borehole pumps.

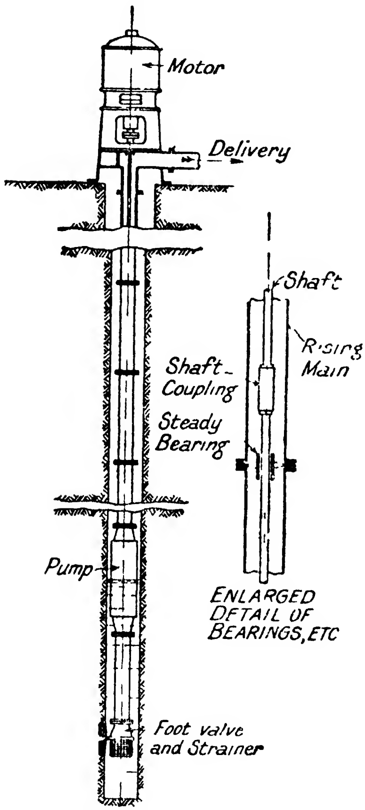


FIG. 89. Shaft-driven borehole pumping plant.

132. Shaft-driven Pumps. Details of a typical installation are given in Figs. 89 and 90. The suction pipe and delivery pipe are co-axial with the pump, and the entire assembly is hung from a water collecting-box or discharge piece at the top of the well. The driving-motor seats itself on the upper face of the collecting-box casting; the driving-shaft which transmits energy to the pump is carried centrally down the rising-main or delivery pipe. In accordance with the custom adopted in this book, the pump and shafting only will here be studied, the upper works being considered as part of the complete installation, § 313.

Pump. The number of stages may vary from two to twenty or more ; only for shallow bores will one stage suffice. If low specific speed centrifugal impellers will serve, then the stages

may be of the ring type, identical with those shown in Fig. 83, § 121; for higher specific speeds the mixed-flow type is preferable, as in Fig. 90 (i). The *rotors* and *diffuser rings* will almost certainly be of bronze. The main *bearings* will be lined either with special self-lubricating bronze, or with lignum vitæ strips. Axial *thrust* is taken not by a balance-disc as in

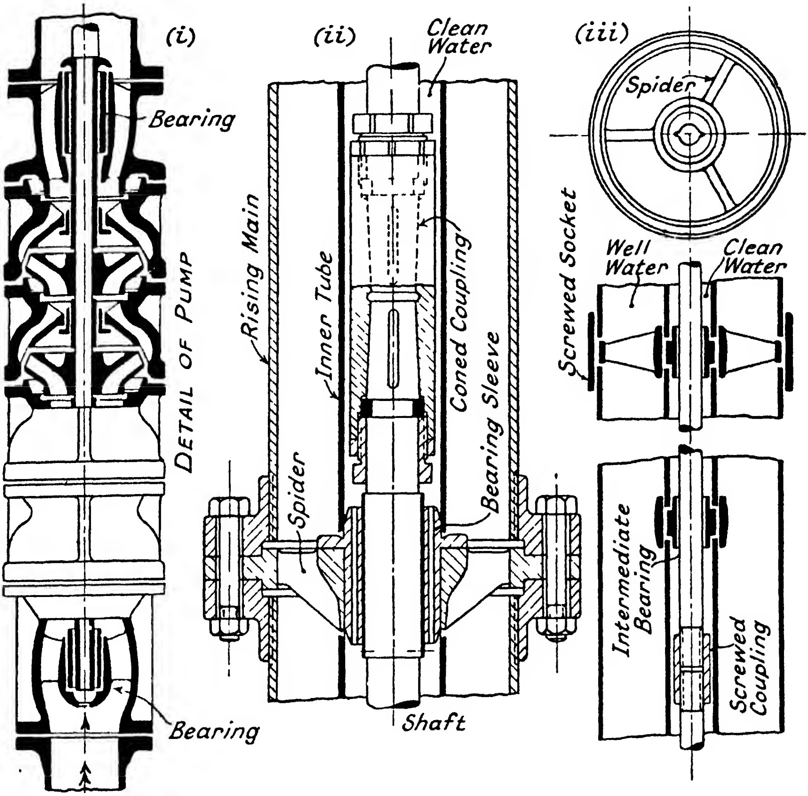


FIG. 90.—Details of shaft-driven borehole pump.

standard horizontal pumps, § 123, but is carried by a thrust-bearing at the top of the borehole. In the pump itself, then, no special provision is necessary.

Rising Main. The vertical delivery pipe has three duties: (i) to conduct water from the pump to the upper collecting box, (ii) to sustain the weight of the pump and the suction pipe, (iii) to support the intermediate guide-bearings for the co-axial

driving-shaft. Now to permit the rapid assembling of these various components it is essential that the individual lengths of the pipe should match the lengths of the shaft, and it is thus the requirements of the shafting that fix the length of each section of the pipe, viz., about 6 to 10 ft. For pipe diameters over about 6 in., flanged joints are suitable ; for smaller bores, socketted joints are better. There are hydraulic as well as constructional reasons for this. The difficulty of getting the water *out of* small bores, § 130, extends to the pipe as well, and that is why we are glad to discard projecting flanges for small pipes and thus use a waterway nearly as big as the bore itself.

133. The Driving-shaft and Bearings. The spacing of the guide-bearings, which in turn controls the lengths of the shaft sections and of the pipe sections, is determined by the resistance of the shaft against whirling. This again depends upon the speed and diameter of the shaft. On the whole it is desirable to run the shaft normally at a speed below its lower critical speed.

The *couplings* for the shaft sections are usually of coned sleeve form, of which a typical example is shown in Fig. 90 (ii). The tapered and keyed end of the shaft is pulled home into the sleeve by an inner screwed bush which bears against a split collar inserted in a groove in the shaft. Lock-nuts make everything secure. Pumps which have screwed rising-mains may have screwed couplings also, Fig. 90 (iii), provided there is no possibility of the shaft accidentally running backwards way.

The housings of the *guide-bearings* take the form of cast-iron spiders clamped between the main flanges of the pipe sections. The bearing shells may be lined with lignum vitæ strips, with self-lubricating special bronze, or with hard rubber. If it is foreseen that much sand will be drawn into the pump, then the entire driving-shaft with its bearings may be enclosed in an *inner tube*, Fig. 90 (ii), (iii), which receives at its upper end a steady supply of filtered water. The inlet pressure is always such as to ensure that leakage will be outwards into the raw water in the rising main, rather than in the reverse direction. Grooves in the bearing-sleeves or openings in the housings allow the clear water to pass from bearing to bearing. Occasionally oil instead of filtered water may be fed to the

guide-bearings, either by filling the tube with oil or by a drip-feed arrangement. Naturally this expedient would hardly be wise if the pump handled drinking-water.

134. Submersible Pumps. In a submersible unit the cost of the electric motor may amount to more than 80 per cent. of the cost of the complete assembly; and certainly the motor has absorbed at least that proportion of the total quantity of ingenuity expended on the problem (*). It has been spent, moreover, to such good effect that the engineer's very natural apprehensions, § 131, have been completely soothed, enabling him at last to share the electrical engineer's full confidence in his under-water motor. Details of the motor itself lie outside the scope of such a book as this one; yet it would show an unfeeling lack of professional interest not to enquire how the electrical designer had solved his very difficult problem. First of all he had to define it: he had to exclude all D.C. motors, all commutator motors, and to concentrate wholly upon the simplest type of A.C. induction motor. The fundamental problem—how to keep the water away from the motor windings—had to be separated into a major part and a minor part. The major part concerned the stator. These windings will undoubtedly be subjected to full supply-mains voltage, and the question of insulation may require quite special study. The minor problem of the rotor conductors which only carry an induced current might admit of a simpler solution. Superadded to this strictly electrical enquiry were those other more or less mechanical questions familiar also to the pump designer—how to counteract the restrictions of the borehole diameter, and how to protect the working parts from abrasion by sandy water. There remained the matter of heat dissipation.

One successful solution has been to exclude water entirely from the motor by enclosing it in a water-tight diving-bell. Although the motor is technically submerged, the water cannot get at it. Thus the insulation need be no different from what usually suffices for very damp localities. Next we come to motors that do actually work in water. The stator coils may now be completely encased in an inner and outer metallic shell, making the whole thoroughly waterproof; but this construction has the disadvantage that the eddy-currents induced in the inner shell may seriously lower the motor efficiency. Using

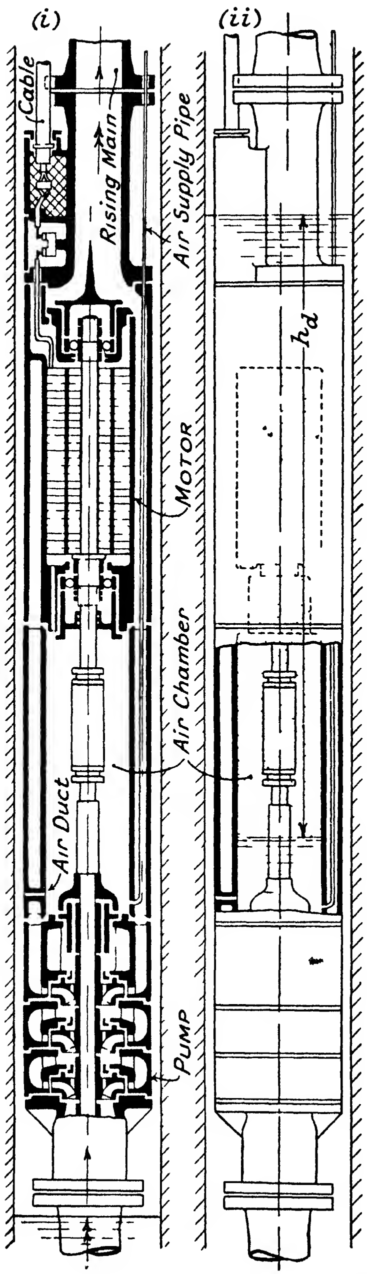


FIG 91 —Dry motor type of submersible borehole pump.

very thin metal—say stainless steel only 0·1 to 0·3 mm. thick —may help, but it is still better to substitute a shell of laminated construction, built up of insulated stampings similarly to the rotor itself. Some motor makers eliminate the inner tube altogether, relying wholly on rubber or similar insulation.

To protect the rotor and its bearings from abrasive water, they may be enclosed in a chamber in which a lubricating fluid (water-oil) continuously circulates ; or a filter incorporated in the casing may admit a slow stream of well water. By these means effective cooling of the motor is also ensured.

For general purposes, then, submersible pumping sets may be divided into (i) *dry-motor* sets, and (ii) *wet-motor* sets.

135. Dry-motor Submersible Pump. As the diagrams show, Fig.91, a typical dry-motor pumping set has the appearance of a long narrow tube of uniform diameter, so proportioned as just to slide down the borehole. The multi-stage pump forms the lower part of the assembly, then there

is a capacious air chamber and finally, at the top, the A.C. induction motor. Water enters the pump normally through a ertical suction pipe, and after leaving the pump the water traverses the narrow annular space between the outer shell and the air chamber or diving-bell. It then flows to the ground surface through a rising-main just as in a shaft-driven set, and this vertical pipe similarly supports the weight of the set. When the borehole water level is at its lowest, atmospheric air has free entry to the air chamber through the duct shown in Fig. 91 (i); and as the motor occupies the upper part of the air chamber, the motor behaves exactly as if it were at ground level. Although the underground motor is certainly deprived of actual air *circulation* or ventilation, it has the compensating advantage of excellent water-cooling furnished by the external water jacket which virtually forms part of the delivery pipe (*).

Suppose now that the borehole water level gradually rises, due to changes in the adjacent subsoil conditions. Eventually the water will overtop the air duct and will begin to enter the air chamber of the pumping set, Fig. 91 (ii). Further rising of the water will progressively compress the air in the chamber, as shown by stippling in the diagram ; but no matter how high the water mounts outside the pump, we can always prevent the water inside from reaching the motor. It is merely a question of making the air chamber capacious enough. After all, the air space in the part of the chamber occupied by the motor itself is relatively small ; so that if we know the limiting pressure-head difference h_d, Fig. 91 (ii), we can quickly work out the corresponding air pressure, the air (compressed) volume, and in turn the free air volume.

If it were not for the possibility of air leakage and the certainty that the water would gradually dissolve the air in the air chamber, the motor would be indefinitely protected. But in fact there must be some provision for periodically renewing the air. Usually a small compressor at the top of the borehole sends down air through a small pipe, a pressure-operated switching device starting the set when required. An alternative system is to fit a water-ring air-compressor, § 153 (*b*), in the pumping unit itself, which draws its air through the pipe passing down the bore.

136. Wet-motor Submersible Pump. "Wet" motors are

always hung below the pump, Fig. 92, and in consequence the direct connection between the pump and the delivery pipe can resemble what is customary for shaft-driven pumps, Fig. 90 (i). A material advantage as compared with the dry-motor type is that no stuffing-box or the like is required at the upper end of the pump shaft; this sensibly reduces design and maintenance problems. Because water now enters the pump through a circumferential strainer fixed between the pump and the motor, there is no need for a suction pipe and the outfit must therefore be set at least several feet *below* the lowest borehole water level (✱).

General comparative comments relating to both types of submersible pumps are :—

(i) *Bearings.* If grease-packed ball or roller bearings are preferred for the motor shaft, this means that the pumping-set must periodically be withdrawn from the well (say after a run of several thousand hours) for replenishment of the lubricant.

(ii) *Axial Thrust.* As the pump shaft and motor shaft are now rigidly coupled together, it may be impracticable to use either of the two chief methods of resisting hydraulic thrust on the pump rotors, i.e., a balancing disc, § 123, or a main upper thrust bearing, § 132. Thus if the motor has ball or roller bearings, these must be proportioned so as to sustain not only the weight of the revolving element but the hydraulic thrust as well. It is true that the hydraulic thrust can be reduced by the use of balancing holes in the rotor discs, § 77, and these are in fact visible in Fig. 91.

On the other hand, if the motor shaft runs in sleeve bearings it will be free to move axially and therefore the pump may have a normal type of balancing disc, as in Fig. 92. The leakage water from the balancing chamber passes through a port back into the well. In the type of motor whose rotor works in a special lubricating liquid, § 134, the auxiliary impellers that circulate this liquid may force it past a balancing disc located beneath the motor. Again, motors using filtered water may embody a water-lubricated Michell type of thrust bearing.

(iii) *Electrical Connections.* Manifestly the question of sealing into the connection-box the insulated cable that brings power down to the pumping set will require quite particular care.

(iv) *Dry v. Wet Motor?* It will now be apparent that for given conditions the wet motor may have a bigger effective diameter than the dry motor. But in the dry motor there are neither the hydraulic frictional losses due to the rotation of the rotor in water or similar liquid, nor the electric eddy-current losses due to the stator protective sleeve, § 134, if this is used.

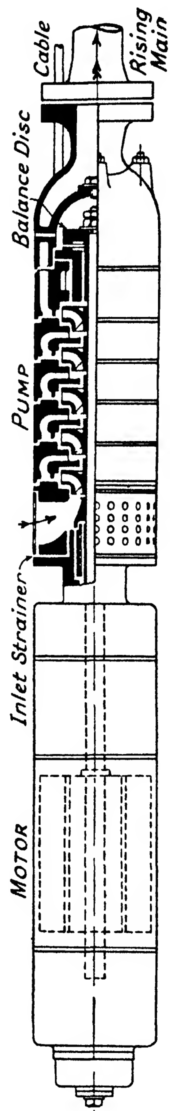


Fig 92.—Wet motor type of submersible pump.

137. Design Data for Borehole Pumps. In settling the main outlines of the pump, relevant matters are :—

Speed. Submersible motors are nearly always bipolar, and thus if the set is supplied from a 50-cycle circuit the pump will run at 2900-2950 r.p.m. Only if the power exceeds about 50 kW. is it likely that there will be room for a 4-pole 1450 r.p.m. motor. A very common speed for *shaft-driven* pumps is about 1000 r.p.m.; the upper and lower limits for these pumps are of the order of 1450 r.p.m. and 750 r.p.m.

Discharge. If, having been given the shaft speed, we now want to find out what size and type of pump rotor will extract a specified amount of water from a borehole of given diameter, then a suitable procedure is :—

For each shape number, or specific speed, there is a fairly rigid connection between rotor diameter and pump casing diameter, § 125. Similarly we may allow a fixed ratio between casing outside diameter and borehole internal diameter, which results in a useful link between rotor diameter and borehole diameter. Now the rotor diameter is itself related to speed and discharge by the empirical rules offered in §§ 92, 106, which finally give this expression :—

$$Q - K_{bh} N D_{bh}^3$$

where Q is the yield from the selected type of rotor, K_{bh} is a constant depending upon rotor specific speed, N is the shaft speed in revolutions per minute, and D_{bh} is the borehole diameter.

If Q is expressed in cubic feet per second (litres/sec.),
D_{bh} ,, ,, ,, ,, feet (decimetres),

then the relation between *shape number* and factor K_{bh} can be roughly estimated, on a conservative basis, as follows : –

Shape number of rotor . .	60	100	150
Value of K_{bh}	0·00017	0·00030	0·00075

The subjoined table gives an impression of how the rotor shape number and the borehole diameter affect the probable yield in gallons per minute, *for a shaft speed of* 960 *r.p.m.*

Shape Number.	Diameter of Borehole (inches).				Speed 960 r p.m.
	10 in.	20 in.	30 in.	40 in.	
60	35	280	960	2,300	Discharge in gallons per minute
100	60	500	1,700	4,000	
150	155	1,250	4,200	10,000	

From information such as this, one could select a suitable rotor type, and finally work out the rotor diameter and head per stage. It is unlikely that the shape number will be less than 60 or more than 200.

Head. It will be noticed that the total head on the pump has not entered at all into the foregoing computations. Only at the end of them do we arrive at the permissible head *per stage*, so that now it only remains to calculate how many stages will be needed to generate the desired head, § 115. This total head may be as much as 800 ft.

Efficiency. The very varied conditions to which borehole pumps are subject make it difficult to give precise estimates of efficiency. Since we are here concerned with the pump itself, rather than with the complete installation, it may be preferable in the first instance to assess the pump efficiency as described

in § 125, and then to make a separate allowance for energy losses in the rising main and in the shaft bearings, § 316.

PUMPS FOR HOT AND VOLATILE LIQUIDS, ETC., ETC.

138. Effects of Temperature Changes, etc. When hot liquids (*) flow through centrifugal pumps they bring with them various new complexities such as these :—

(i) Thermal expansion and contraction of the parts of the pump must be carefully studied, and suitable provision made against risk of damage from this cause.

(ii) High temperatures may so greatly reduce the viscosity of the liquid that leakage may now occur through joints in the pump casing that were ordinarily liquid-tight. New jointing materials or methods may be essential.

(iii) Liquids which are innocuous at normal temperatures may grow aggressive at high temperatures and may corrode elements of the pump which were formerly judged quite resistant. Different materials of construction will in these instances be required.

(iv) The main stuffing-box of the pump requires competent maintenance even when subjected to low pressures and atmospheric temperatures, § 86. When there is a combination of high pressure, high speed, and high temperature, only a fundamentally new design of stuffing-box may meet the conditions.

(v) Hot liquids and volatile liquids have an increased tendency towards vaporisation. If the pressure is suddenly reduced, the liquid may "flash" into vapour. The pump must be so constructed and operated as to prevent such flashes. In a similar way the increase in vapour tension as temperatures increase will materially affect the permissible suction lift on the pump: the pump may even have to be set *below* the suction-well level, § 250.

(vi) Alterations in liquid density and viscosity such as accompany temperature changes will have a marked influence on the pump performance, § 255. Thus, if the rate of flow and the shaft speed were kept unaltered, a pump generating 1000 lb./sq. in. with water at 50° F. would generate a pressure of only 860 lb./sq. in. at 400° F.

139. Expansion: Leakage. Although a pump when heated up should expand uniformly in all three directions, we may look for trouble chiefly from vertical expansion and longitudinal expansion. Consider, for instance, a multi-stage pump supported on feet disposed normally beneath the body, Figs. 82, 83. During erection it would be an easy matter to bring its axis exactly in line with the shaft of the driving-motor. But in use, if the liquid—e.g. oil—had a temperature of 500° F., the vertical expansion of the pump body might lift the shaft out of line with the motor shaft by an amount beyond the range of accommodation of the flexible coupling. Similarly the longitudinal expansion might distort the casing or even break off the feet. Preventive measures adapted to a barrel-type multi-stage pump, § 121, are shown in Fig. 93. The casing, in this instance of forged steel with welded branches, is supported by welded lugs lying in the horizontal plane coincident with the axis; and in order to make sure that the bedplate itself suffers no change of shape, the stools on which the lugs rest may themselves be water-cooled. Longitudinal expansion of the casing is freely allowed, for the supporting lugs are only secured at one end; at the other end they may slide over the bedplate stools, being constrained sideways only by longitudinal keys.

Another possibility of trouble concerns the shaft and impellers. If dissimilar metals are used for these components, differential expansion may result. The impellers, which appeared to be tight on the shaft at workshop temperatures, may develop slackness at high pumping temperatures.

Split-casing high pressure pumps, § 120, are regarded by some users as unsuited for very high temperatures; the main longitudinal joint is likely to be too troublesome. But the ring type of pump, § 121, gives good service with extremes of temperature and pressure. The general style of mounting resembles what is sketched in Fig. 93, and in addition the main longitudinal tie-bolts are given special care. In one patented form of construction, they are enclosed in a casing through which water at a controlled temperature is circulated, thus ensuring that they are not unduly strained if the pump is too rapidly heated up. This outer casing also simplifies the arrangement of the lagging or heat insulation that such a pump

will certainly need. In regard to the circumferential joints between the diffuser rings, many makers claim that these do not leak if of suitable metal-to-metal construction.

140. Heat-resisting Materials. It is worth remembering that at least where water is concerned, high temperatures and high pressures inevitably go together—otherwise the water would vaporise. Experience has shown that for the respective components of the high-temperature pump, the following range of metals is appropriate :—

Casing. Forged steel or cast steel will alone meet the conditions. Single-stage pumps may have the casing machined from a solid block of steel.

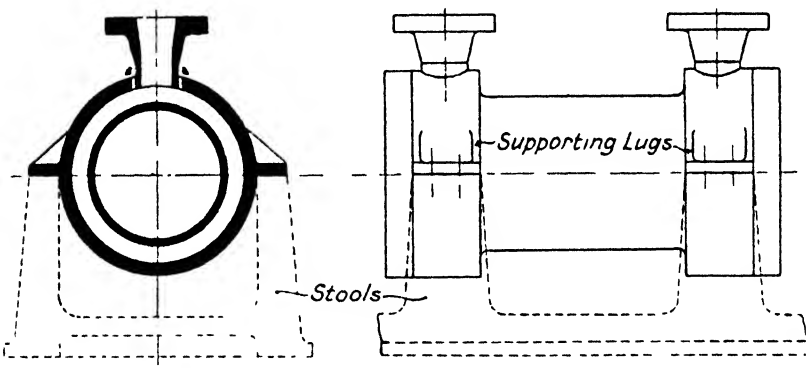


FIG. 93. Method of mounting casing of multi-stage pump handling hot liquid.

Shaft. Forged steel, chrome-nickel steel, or nickel-chrome-vanadium steel.

Impellers and Diffusers. Monel metal, cast stainless steel, or copper-nickel alloy.

For boiler-feed pumps it has been suggested that the *p*H value (hydrogen-ion concentration) of the water may give useful guidance (*). For example if the *p*H value is below 3·5, the working parts should be of corrosion-resisting steel, while bronze would be suitable for a range of *p*H values of 3·5 to 6·0. Between 6·0 and 8·0, there might be a choice of bronze, iron, or gun-metal, while an all-iron or all-steel pump would serve for still higher *p*H values. Manifestly these are provisional suggestions only, to be accepted in conjunction with other characteristics of the feed-water.

141. Stuffing-boxes. In the special types of stuffing-box that high-speed, high-pressure, high-temperature pumps may require, it may be necessary to make provision for any or all of the following needs :—

(i) The packing may require protection against excessive pressure.
(ii) An external jacket may cool the packing.
(iii) Sealing, cooling, or lubricating liquid may be circulated internally.

Figs. 94 to 96 show in schematic form a few suitable methods (*).

Pressure-reducing Systems. Soft packings must not be subjected to excessive pressures for the same reason that they must not be squeezed too tightly by injudicious screwing-up of the gland nuts, § 81 ; the packing might be forced so hard laterally against the shaft that overheating and rapid wear would ensue. In standard pumps it is often an easy matter to ensure that high-pressure liquid never comes near the packing, even though the pump generates high pressures, § 124. But now we must constantly keep in mind the association of high temperature and high pressure : no matter whether the shaft emerges from the pump casing at the high-pressure end or at the (nominally) low-pressure end, the pressure-difference between the internal liquid and the external atmosphere may be very considerable. If this pressure-difference is more than we consider the packing can support, viz., more than 200 lb./sq. in. or so, how can we dissipate the redundant pressure ?

Fundamentally there is only one way. We must allow the liquid to leak through a narrow opening and thus destroy its energy in hydraulic friction or in turbulence. Examples of this technique have already been mentioned in connection with wearing-rings, § 83. and balancing-discs, § 124 ; new applications are illustrated in Fig. 94.

142. Packing Protection, (a) Pressure. In Fig. 94 (i) a stuffing-box is shown having a bushing of length l interposed between the pump casing and the packing. From a small intermediate chamber, leakage liquid can be drawn off through an adjustable valve. When the valve is closed there will be no pressure-drop in the bushing, and thus the first packing-ring will be fully exposed to the local pump pressure p. But if the

valve is opened slightly so as to permit flow through the narrow annular clearance space between bushing and the shaft, the resulting pressure-drop p_l will reduce the pressure actually operating on the packing to the value p_p, which may be well within safe limits. If the required drop cannot be created without wasting too much liquid or using an excessively long bushing, then a double serrated sleeve (ii) or a labyrinth, Fig. 94 (iii), may be preferred.

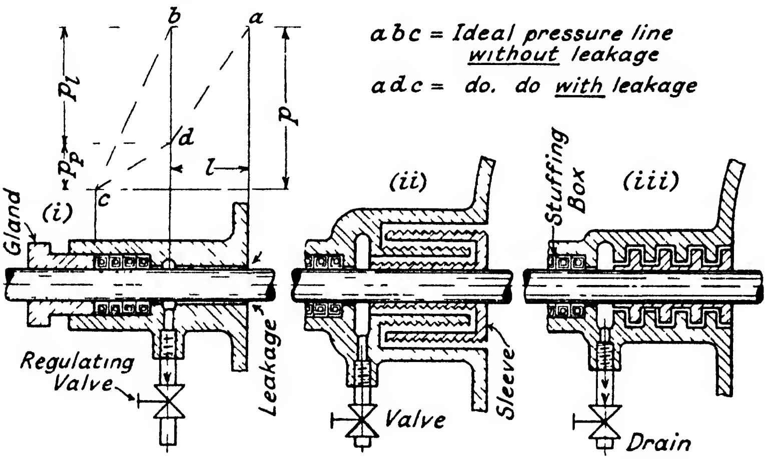


FIG. 94 Diagrams showing leak-off systems for reducing pressure on stuffing boxes

How to dispose of the leakage liquid is a matter that can only be decided when the layout of the installation as a whole is settled. There is now an additional reason for choosing the most favourable point for returning the liquid to the main circuit. Formerly all such liquids only carried away hydraulic energy, § 192 ; but now they carry heat energy as well. Consequently a region should be sought where the temperature and pressure differ as little as possible from those prevailing at the point where the leakage emerges from the regulating-valve beneath the stuffing-box. (If this valve is omitted altogether, there will be no danger of leaving it accidentally closed.)

143. Packing Protection, (b) Temperature, etc. *Cooling* of the stuffing-box not only protects the packing, but to some extent it serves to reduce at least slightly the heat conducted

along the shaft to the main bearings (*). A plain water-jacket for the packing is illustrated in Fig. 95 (i), while at (ii) the jacket surrounds the shaft and is thus more efficacious in withdrawing heat. By the time hot liquid from the pump casing has penetrated along the narrow clearance space, it should have little power to harm the packing. If a leak-off as indicated by broken lines is provided from the intermediate chamber, then pressure-reduction as well can be effected, as in Fig. 94 (i). A combination of systems (i) and (ii), results in the disposition Fig. 95 (iii), in which the jacket cooling water may be directed over circumferential cooling ribs. Alternative positions for the cooling-water inlet are here shown.

Internal cooling can be contrived by forming an annular space between the shaft wearing-sleeve, § 80, and a part of the

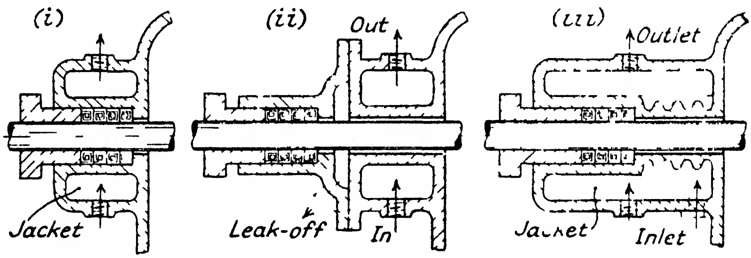


FIG. 95 Diagrams showing cooling systems for stuffing-boxes etc

shaft that is reduced in diameter, Fig. 96 (ii). Cooling liquid introduced into this space through a longitudinal hole escapes through a chambered recess in the gland. An alternative method of supplying cooling liquid to the annular space is by a pair of lantern rings, § 85 (ii), one at either end of the stuffing-box.

Sealing of the stuffing-box can be effected by using a lantern ring, just as it was in a standard pump. But now the conditions are different. It is not a question of keeping air out of the pump, but rather a question of keeping the liquid in the pump from escaping into the air. High-pressure water, for example, would flash into steam, while spirits might gasify and form explosive mixtures with the air. A continuous stream of liquid through the lantern-ring, Fig. 96 (i), will certainly help to cool the packing, and if the liquid is properly chosen it will lubricate the shaft as well. External jacket-cooling is additionally provided for both of the systems shown in Fig. 96.

The pressure at which sealing or cooling liquid is supplied to the lantern ring may require careful regulation. If it is inadmissible for the sealing liquid to contaminate the pumped liquid, then clearly the sealing liquid must be kept at the lower pressure of the two. On the other hand, if there is no objection to a small amount of lubricant penetrating into the pump, then a supply of lubricating liquid may be used for sealing purposes and circulated continuously; its relatively higher pressure would protect it from admixture with the pumped liquid. But as, in any event, there will certainly be leakage of *sealing* liquid from the gland into the atmosphere, the liquid must itself be innocuous.

144. Types of Packing. For use in the special types of stuffing-box just described, we can hardly expect that the soft

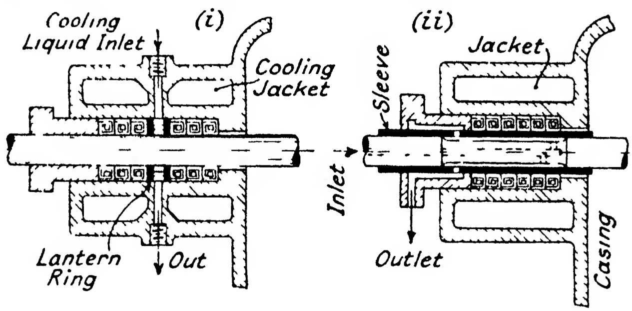


FIG. 96 Diagrams showing internal cooling of gland packing, etc

cotton packing that serves so well for cold-water pumps, § 86, will still remain serviceable Instead there is a range of suitable heat-resisting materials available, which includes asbestos either alone, or reinforced with rubber, "plastic" compounds, or even with metal foil (lead, copper, aluminium). If one could count on the pressure gradient in the stuffing-box obeying the ideal law, *adc*, Fig. 94 (i), then it might be advantageous to use in different parts of the box different packings to suit the local pressure. Thus at the high pressure end of the stuffing-box, relatively hard rings would probably be obligatory to meet the high local pressure; while at the low-pressure end one might hope that softer rings would serve. Such differentiated packings have given good service in practice. Another solution is to rely wholly on rotating metallic sealing-rings, kept in place by springs and properly lubricated.

Hard packing-rings subjected to high-pressure would certainly cut and score bronze shaft-sleeves such as serve for cold-water pumps, § 80. For high temperature pumps, only hardened and polished steel sleeves will resist the tendency to rapid wear. Finally, it cannot be forgotten that a shaft that runs as truly as possible and is free from all vibration is the basic requirement for satisfactory service in these severe conditions.

145. Condensate-extraction Pumps. The water extracted from surface condensers is admittedly not very hot, but neither is it cold. In any event the close collaboration between boiler-feed pumps, and the machines which deliver water to them, entitles condensate-extraction pumps to a place in this chapter ; and besides, the problems of sealing that earlier paragraphs have just examined are now of overmastering

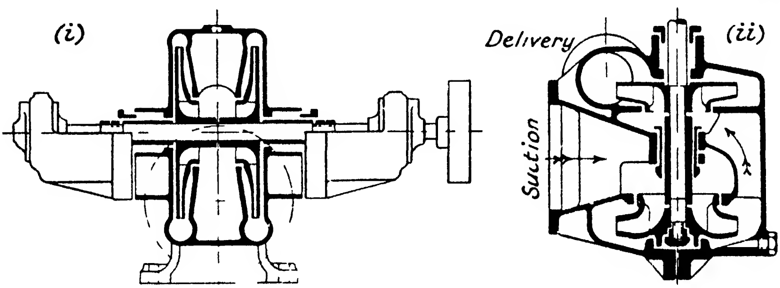


FIG. 97 Types of condensate extraction pump

importance. The whole design of the pump is controlled by the necessity for excluding any air whatsoever. The pumps are intended for use in a closed circuit in which the most extraordinary precautions are worth while to prevent air reaching the boiler. Condensate-extraction pumps also naturally work under high vacuum heads, and as the pump must be set as small a distance as possible below the condenser, § 344, energy losses in the inlet branch of the casing must be as small as possible.

Air Exclusion. This can most effectively be ensured by using two impellers set eye to eye, Fig. 97. If the main suction branch communicates with the space between them, it necessarily follows that the stuffing-boxes are on the outside or *delivery* side of the impellers. As the absolute pressure here is greater than atmospheric pressure, air cannot possibly get past

the glands. Sometimes the twin single-inlet impellers work in parallel, Fig. 97 (i), and sometimes a two-stage arrangement is preferred, (ii). The shaft may be either horizontal or vertical.

Although in either event a split-casing disposition is convenient, yet it introduces the risk of air leaking in along the line of the joint between the two halves of the casing. To eliminate this danger, a groove or channel may be formed the whole way round the joint, this annular space being filled with water at delivery pressure. In this way the suction space of the pump is everywhere guarded either by solid metal or by a continuous barrier of high-pressure water.

High Vacuum. This requirement is fulfilled by modifying the shape of the impeller and by following the rules laid down in § 255 (iii) *b*; such changes may sensibly lower the pump efficiency.

Low Inlet Loss. The casing inlet branch is made exceptionally large—perhaps having a diameter more than double that of the outlet branch, Fig. 97.

Pumps for Viscous, Corrosive, Abrasive Liquids, Etc.

146. Range of Material to be Pumped. From the point of view of the pump, liquids can be graded according to a continuously rising scale of impurity, viscosity, or corrosiveness, ranging all the way from distilled water on the one hand to thick mud, dredgings, or concentrated acid on the other. Already we have made a little progress along this path. Thus a nominally standard pump may require slight modification to fit it for handling the slightly acid water of mines, § 129, or for resisting the slightly aggressive action of hot water or hot oil, § 138. Similarly the nominally clean cold water flowing through an irrigation pump may in fact be charged with quite appreciable amounts of sand or silt.

But now the pump may have to contend with far heavier degrees of contamination; in brief, it must be able to handle *anything that will flow through a pipe*. That is why the title of this paragraph mentions "material" rather than liquid. The substance that passes through the pump may not be worthy of the name liquid at all. Indeed the pump's prime duty may be to transport *solids*, and not liquids, the solids being in

granular form so that when mixed with water they form a sludge or slurry. The water serves purely as a carrier, as it does in, e.g., water-borne sewage.

In general, then, one might make a very rough division of the duties now in question as follows :—

Viscous or Corrosive Liquids relatively free from Suspended Solids

Thick oils.

Industrial and other products such as fruit-juice, milk, beer, liquors, etc., etc.

Chemical products such as acids, alkaline solutions, etc., etc.

Liquids charged with Floating or Suspended Material

The solid and semi-solid material may include :—

In the paper industry : rags, wood pulp, esparto grass, etc.

In metallurgical processes : ground-up ores and minerals of many kinds.

In general :—

Sewage, e.g., paper, rags, refuse, etc.

Leaves, weeds, straw, chips, etc.

Coal dust, ashes, flue dust, clinker, cement sand, mud, silt, grit, pebbles small rocks, etc.

Offal, grease, fat, etc.

Miscellaneous matter in trade wastes, e g., from sugar-beet works, etc.

147. Modifications in Pump Design. According to the nature of the liquid or the fluid mixture (*), the designer may have to make special provision against

(i) The chemical or electrolytic corrosive effect of the liquid on the internal parts of the pump.

(ii) The abrasive or erosive action of the suspended solids.

(iii) The tendency of the larger solids to clog or choke the passages of rotor or casing.

(iv) The release of abnormal quantities of entrained air or gas.

In regard to items (i) and (ii), we may either make the pump internal parts from very hard or resistant materials, or we may use an ordinary material such as cast iron and protect it by an inner resistant lining. Such resistant materials include :—

Bronze.
Special cast iron, e.g., silicon cast iron.
Steel : manganese steel, chrome nickel steel, cast steel, stainless steel.
Lead.
Stoneware, porcelain, glass.
Hard rubber, etc., etc.

In regard to item (iii), all internal projections or constructional elements likely to obstruct free movement of the solids must be eliminated from the pump design. These include fixed guide-vanes of any kind, internal bearing housings, etc. The rotor also may need radical modification in shape. Hand-holes with quick-opening covers will be desirable, to permit accumulations of solids to be cleared from the casing without waste of time. Similarly if loose renewable linings are provided, provision should be made so that new ones can be quickly exchanged for worn ones.

Additional questions to be studied include :—

(*a*) Unless the design of the stuffing-box is altered, the abrasive solids may rapidly score the shaft or shaft sleeves.

(*b*) A special supply of filtered water or liquid may be needed for the lantern-rings, § 85.

(*c*) Hot-water jackets for the casing may be required to keep thick liquids sufficiently fluid to pass through the pump.

(*d*) Cutting appliances inside the pump may cut up fibrous solids that would otherwise choke the rotor passages.

In any event the primary need is for a general stiffening of the whole construction. This is because we can no longer rely on uniformity of flow through the pump. The floating or suspended material may not come along at a regular rate ; it will probably arrive in sudden gulps. In consequence violent and unpredictable unbalanced forces may fall upon the impeller and shaft, and the resulting risk of dangerous vibration can only be countered by skilful disposition of additional metal.

The *efficiency* of the pump inevitably suffers : if the designer is so much preoccupied with his task of shepherding unpromising mixtures of material through the pump, he is obliged to let other matters take their chance. At worst, the gross efficiency may fall as low as 50 per cent. The greater the

departure from the standard rules of design set out in Chapter VII, the lower the efficiency is likely to be.

148. Sub-standard Pumps. This term is intended to describe machines whose general design does not seriously depart from normal. Examples are :—

(i) *Sea-water Pump.* Only new materials of construction are required here, the general shape of the passages remaining unaltered. Non-ferrous metal, e.g., bronze, gunmetal, etc., should preferably be used throughout, alike for the impeller, the casing, and the shaft. Only in this way can the danger be averted of electrolytic action at the junction between ferrous and non-ferrous metals.

(ii) *Stuff Pump.* In the paper industry the concentration of rags, pulp, etc., may often be less than 2 per cent., and in that event relatively minor alterations will permit standard designs to serve (*). Split-casing double inlet pumps are suitable, Fig. 39 (ii), except that open-type impellers as in Fig. 37 (IV) are substituted. As the wear on the sides of the casing adjacent to the blade edges may be heavy, renewable side plates as seen in Fig. 99 may be advantageous.

(iii) *Oil Pumps.* Machines that are virtually standard will serve for a wide range of oils and petroleum products. But for high-temperature service in refineries, the pump must naturally be modified as described in §§ 138 144.

149. Unchokeable Pumps. This class of machine represents the limit of the range ; its design is now wholly subordinated to the requirement that *anything that will pass through the suction branch will also pass through the rotor passages.* Naturally the stipulation must not be taken too literally ; it means in effect that solids up to 6 in. in diameter may be expected, for even if the pump branches are much bigger than that, the water could not convey anything larger. It is also the transporting power of the water that sets a lower limit to the size of unchokeable pumps ; if the pipes and branches were less than about 3 in. in diameter, the corresponding water velocity would not be high enough to keep the pipes clear.

Almost invariably *side-inlet impellers* are preferred, usually of the open type, Fig. 37 (III), but sometimes of the shrouded type (I). The number of blades is hardly ever more than *four*, and is quite often only *two*. Fig. 98 shows what great changes

from standard blade form are essential; type (i) is a patented shape intended to resist clogging by stringy or fibrous material, while type (ii) is suited to a wide range of liquid-solid mixtures. Comparing Figs. 37 and 98, we notice that the projecting boss of the standard impeller is now almost eliminated.

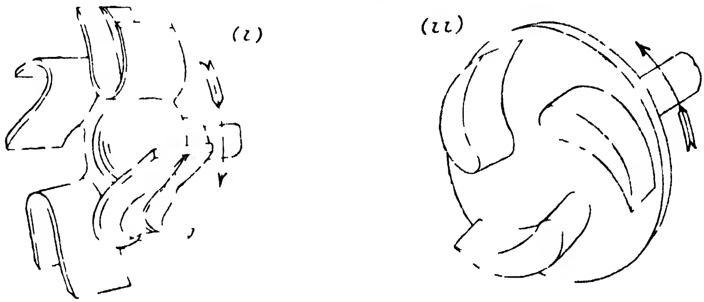


FIG. 98 Impellers for unchokeable pumps

The unchokeable pump illustrated in Fig. 99 has a shrouded or closed 2-bladed side-inlet impeller, and its suction branch has a hand-hole fitted with quick-opening door which permits the impeller eye to be cleared. Renewable side-plates protect the pump casing and cover against the heavy wear which may be experienced in these regions.

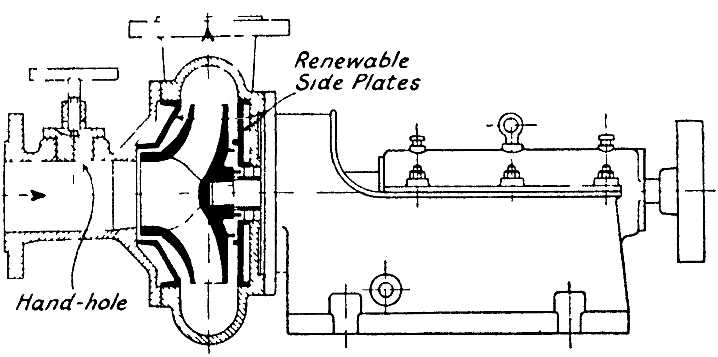


FIG. 99 Unchokeable pump.

150. Lined Pumps. An example of a pump built of standard materials but fitted with resistant linings is shown in Fig. 100. In this instance the lining is of hard rubber, which is found very suitable when the water contains quantities of ashes or grit in suspension (*). The impeller is of the general

pattern depicted in Fig. 98 (ii), and in addition it has special vanes at the back for the purpose described in § 151. At the bottom of the casing there is a cleaning door.

If the lining is of stoneware, glass, porcelain, or the like, intended to resist chemical action, then one of the chief problems of design is to ensure that the vulnerable cast iron parts of the pump are thoroughly protected, even at the joints. Since the slightest unobserved leakage of corrosive liquid might attack and seriously weaken the bolts that fasten together parts of the

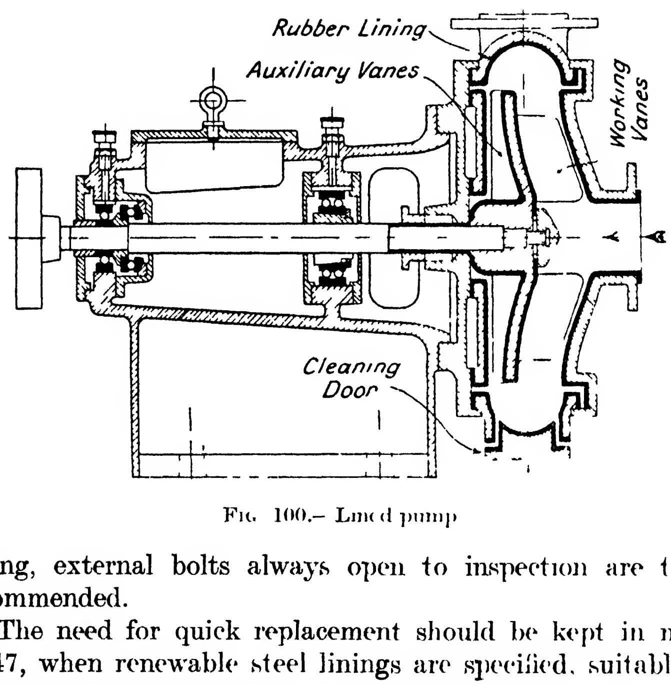


FIG. 100.– Lined pump

casing, external bolts always open to inspection are to be recommended.

The need for quick replacement should be kept in mind, § 147, when renewable steel linings are specified, suitable for use with liquids containing highly abrasive material

151. Axial Thrust: Leakage. In regard to axial thrust, it is clear that if the liquid contains floating or suspended matter we can no longer rely on the balancing holes and sealing ring described in § 77 ; the holes would choke up and the sealing ring would either jam or very rapidly wear. The only alternative is a heavy external ball-thrust bearing, as in Fig. 100.

Normal types of *sealing ring*, as in Fig. 99, can be protected by a supply of clear water fed into the clearance space.

As for the *stuffing-box*, new solutions must be found to the problems posed in §§ 85, 86, or 141. Some possibilities are :—

(i) We can so control the local pressure inside the pump, close to the stuffing-box, that there is the least possible tendency for the gritty or corrosive liquid to leak along the shaft and attack the packing-rings. This will naturally occur when the absolute pressure here corresponds to atmospheric pressure p_a. Now the absolute pressure at the impeller rim, p_r, will almost certainly be higher than this—its value will depend upon the total head on the pump and upon the suction lift, § 163. A method of neutralising this pressure difference $(p_r - p_a)$ was suggested in § 74. In a standard pump it was shown that there already existed a pressure difference of $w\left(\frac{v_2}{2}\right)^2 \Big/ 2g$, Fig. 42 (ii); now we must realise that this could be increased quite considerably by *controlling the speed of revolution* of the liquid in the clearance space between the impeller and casing. Hitherto we have arbitrarily assumed that the speed of the liquid elements is one-half of the impeller speed. If vanes are cast on the *back* of the impeller these would enforce a much higher speed, which in turn implies a bigger pressure difference (*).

It is the radial length of the auxiliary vanes that controls the intensity of the pressure difference. In the pump illustrated in Fig. 100, these vanes (which of course have no functional connection whatever with the working vanes) extend actually beyond the tips of the working vanes.

(ii) The principle of the lantern-ring, § 85, may be applied. If clean water can be allowed to mix with the pumped liquid, then liberal supplies from a separate source can be fed into the stuffing-box, thus washing away from the neighbourhood all gritty particles.

(iii) If the liquid levels and the pressure-control vanes ((i) above) are suitably adjusted, it may be possible to do without a stuffing-box altogether; such leakage as occurs can be taken back to the suction well. Vertical-shaft pumps especially can be so designed.

Other Special Types of Pump

152. Self-priming Pumps. In their usual conditions of service, normal types of rotodynamic pump will not begin to

deliver liquid until they have been "primed", § 291. Before the machine is started, residual air must be removed from the pump casing and suction line, and replaced by liquid. A *self-priming pump* is one that performs this operation automatically; no matter whether air or liquid enters the suction pipe, the apparatus will deal equally well with either.

Although self-priming pumping sets are built in a great variety of types, they may roughly be classified into :—

(i) Dual-service pumps which by their own nature are adapted to work with air or with liquid. Some of them are hardly entitled to the name "rotodynamic pump" at all.

(ii) True centrifugal pumps which have incorporated in them a *priming nozzle* which projects a jet of liquid into the eye of the single-inlet impeller.

(iii) Centrifugal pumps fitted with diffusers, § 44, in which the priming action takes place at the tips of the impeller blades.

(iv) Standard centrifugal pumps working in parallel with an independent air-exhausting pump which automatically comes into action when the main pump is started.

Although the dual-service pump is the simplest in construction, its efficiency is low and it is only preferred when the power input does not exceed a few horse-power. As by any system the self-priming effect can only be bought at the cost of some loss of efficiency, large pumps are rarely so adapted; they have an independent priming system as described in § 292. In addition, there are other possibilities, as explained in §§ 338-340.

153. Dual-service Pumps. (*a*) *Turbine-type.* The term "turbine pump" is sometimes applied to any standard centrifugal pump provided with a diffuser type of recuperator, § 44. Here the term describes a special kind of small pump in which the rotor blading resembles the blading of a steam-turbine rotor; the pump rotor is housed in a casing which again is quite unlike a normal volute or normal diffuser. The inlet and outlet branches are disposed in the top of the casing; consequently if the casing has initially been filled with liquid, it will always remain full after the pump has stopped. On re-starting the pump, internal circulation of the liquid creates a suction effect which quickly exhausts air from the suction pipe,

whereupon liquid begins to flow normally through the circuit. The permissible suction head may be as high as 20-25 ft.

(*b*) *Liquid-ring Pump.* The rotor of this pump, Fig. 101, has radial blades, and it is mounted eccentrically in a circular casing divided by a diametral partition. Fluid enters the rotor through a crescent-shaped port on one side of the partition, and leaves through a similar port on the other side. The action is as follows : Assuming that the necessary initial charge of liquid has been poured into the casing, then when the rotor begins to revolve it drives round with it the liquid caught between the blades. Eventually, after full speed has been reached, centrifugal force acting on the rotating liquid causes it to form a continuous ring which is forced against the outer circumferential wall of the casing—hence the term " liquid-ring pump ".

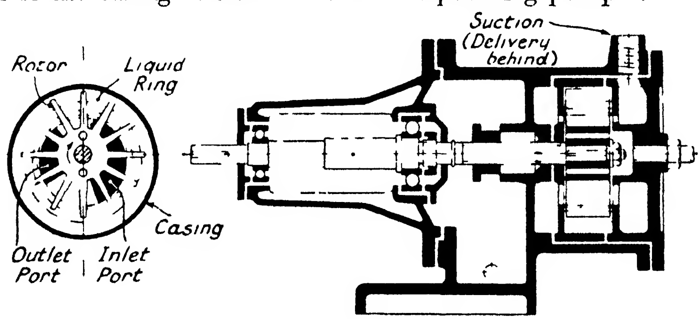


FIG. 101 Water ring pump

Now let us examine the series of small cells that lie within this ring. Each of these cells is enclosed by (*a*) two adjacent rotor blades, (*b*) the inner surface of the liquid ring, (*c*) the parallel sides of the casing. Now because of the eccentricity of the rotor shaft, we observe that as the rotor revolves clockwise, the volume of each cell *constantly changes* ; in the diagram, Fig 101, the volume of the descending cells increases while the volume of the ascending cells diminishes. To occupy the increased space made available in the descending cells, air enters from the pump suction pipe, and this air is discharged into the delivery pipe on the upward journey. In this way the suction system is progressively evacuated. When all the air has been drawn out, liquid now follows ; and it continues to pass through the pump just as the air did. Thus the pump will discharge air or liquid whichever is fed to it (*).

Quite evidently the liquid-ring pump is not a true rotodynamic machine; it is only described here because it is such a useful collaborator with centrifugal pumps.

154. Pumps with Priming Nozzles. In these machines, § 152 (ii), the rapidly-moving impeller blades cut through the priming jet and break up the liquid into a dense cloud of spray which continually sweeps through the impeller passages, entraining with it a steady stream of air from the casing and suction pipe. As soon as evacuation is complete, liquid now flows up the suction pipe and into the impeller, whereupon normal pump operation begins. As in all self-priming pumps, the casing holds a sufficient reserve of liquid to begin the priming process as soon as the shaft begins to turn (*).

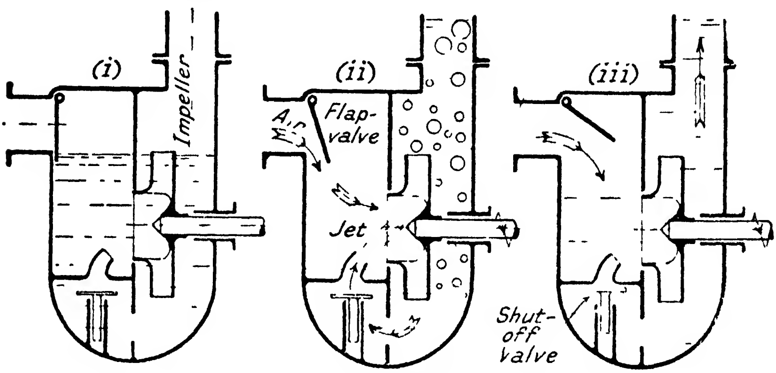


FIG. 102.—Diagrams showing working of self priming pump.

The fully-automatic self-priming pump shown schematically in Fig. 102 has its suction branch arranged in the top of the casing, with a flap-valve serving as a non-return valve, § 290. Another automatic valve cuts the priming nozzle out of action when the pump begins to work normally. Three phases in the starting of the pump are illustrated in the diagram, thus :—

(i) The pump is dead, and the impeller is submerged by the residual charge of liquid. Had it not been for the closed flap-valve, this liquid might have siphoned back down the suction pipe when the pump was last shut down.

(ii) The pump has been started and is in process of evacuating the suction spaces. The revolving impeller has drawn down the liquid in the suction side of the casing and raised it on the delivery side ; the resulting head-difference generates a

continuous flow through the priming nozzle. Of the air-liquid mixture discharged by the impeller, as explained above, the air passes away up the delivery pipe and the liquid is re-circulated. Meantime, air from the suction pipe has pushed open the flap valve, and the vacuum steadily grows stronger; in turn the head-difference acting on the nozzle increases and the action of the priming jet grows more intense.

(iii) Liquid from the well having by this time followed the air up the suction pipe, and all the air having now been evacuated, normal pump operation begins. "Solid" liquid flows through the impeller—which now generates its full working head—and out through the delivery branch. The shunt or by-pass flow through the priming circuit has been so powerful that it has lifted the shut-off valve and, by stopping the flow, has prevented further waste of energy there.

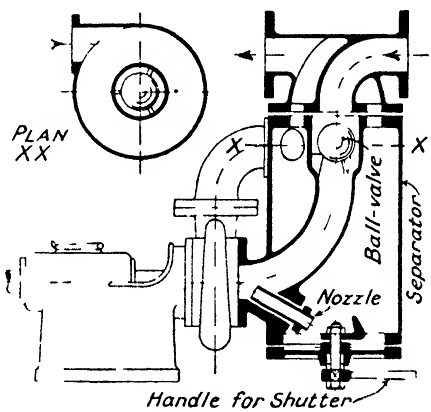


Fig. 103 Standard pump fitted with priming nozzle and air separator.

155. Pump with Improved Priming Circuit. In self-priming pumps fitted with evacuating nozzles, the speed and certainty of the priming operation depends a good deal upon the effective separation of the air from the liquid. This can be encouraged by providing a special separator, as indicated in Fig. 103. The pump is now virtually of standard form; only the inlet and outlet passages are unusual (*) To the suction flange is bolted a vertical cylindrical container which serves both as a reservoir for the supply of priming liquid and also as a centrifugal separator. An axial passage leads air or liquid into the pump, and this passage houses a ball type of reflux-valve. The pump delivery branch is connected to the upper end of the cylindrical container; as shown in the sectional view XX, the connection is arranged tangentially, so ensuring that a rotary motion is imparted to the air-liquid mixture discharged by the

impeller during priming. It is this vortex motion which swiftly separates out the air. The heavier fluid, viz. the liquid, is forced against the cylindrical wall of the container, thereafter descending to the priming nozzle ; the lighter fluid, air, remains near the core of the vortex and can pass freely away up the main delivery pipe.

In this relatively large unit the priming circuit is shut off by a hand-operated shutter, Fig. 103, rather than by an automatic valve as in Fig. 102.

156. Combined Self-priming Pumps. There are various ways of assembling together a standard centrifugal pump and

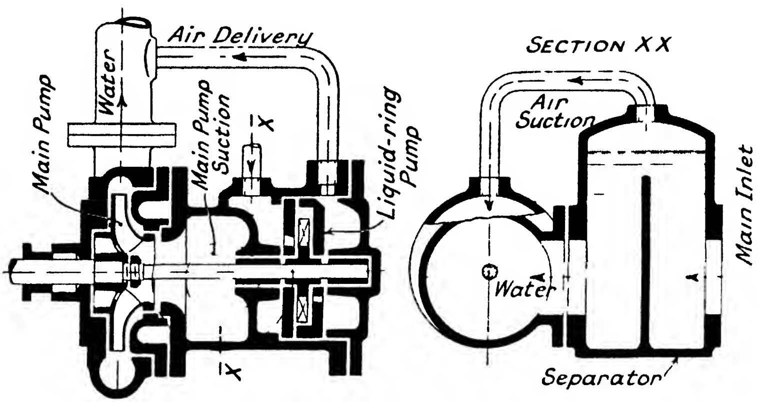


Fig. 104.—Combined self-priming pumping set.

the auxiliary pump which will be called upon to evacuate the entire circuit when the set is started, § 152 (iv).

(*a*) In the system illustrated in Fig. 104, a liquid-ring pump attends to the priming, § 153 (*b*), its rotor being mounted on the same shaft as the centrifugal pump impeller. To the suction chamber of the centrifugal pump there is bolted a separator which delivers liquid to the main pump and air to the priming pump. When the set is started, the impeller revolves idly in air and the priming pump begins to draw air out of the suction pipe ; when " solid " liquid finally reaches the set, the centrifugal pump and the liquid-ring pump share it between them. The two pumps do strictly work in parallel.

(*b*) As the liquid-ring pump is never so efficient as the centrifugal pump, the close-coupled set shown in Fig. 104

always absorbs slightly more energy than an equivalent centrifugal pump alone. In order to overcome this disadvantage, the priming-pump may be mounted on the pump frame as an independent motor-driven unit. Its own driving-motor may be controlled by a pressure-operated switch connected to the main pump delivery pipe. The switch-gear is so arranged that when the main pump motor is started the auxiliary circuit is also energised ; as the main pump impeller is as yet running in air, no pressure is developed and the pressure-switch automatically closes and starts the priming pump motor. When priming is complete and liquid pressure builds up in the delivery pipe, the switch opens, the priming pump stops, and its motor absorbs no further energy.

The vertical pump shown in Fig. 40 (i) can conveniently be made self-priming by mounting above it a liquid-ring pump. Although the priming pump is driven at the same speed as the main pump, the two hydraulic circuits can be kept quite distinct by means of a float-operated valve, which is advantageous if the liquid passing through the main pump is too dirty to be relished by the liquid-ring pump. There is also the possibility of reducing the priming-pump energy consumption during normal running by partially draining liquid from its casing.

157. Hydraulic-storage Pumps. If hydraulic-storage pumps are not always special in construction, they are sometimes unique in size. No other pumps yet built will transmit such large amounts of energy. Units of nearly 40,000 h.p. each are in operation, and units of more than 60,000 h.p. are projected. Indeed we are more interested in the energy than in the water, for the water is merely the medium by which excess energy from another source can be utilised and stored in hydraulic storage basins, § 349.

A constructional peculiarity sometimes imposed by the severe operating conditions the pumps must meet is indicated in Fig. 105. The internal stay-vanes, § 88 (ii), are composite, each having a fixed part and a pivoted moveable part. By means of an external system of cranks and links, all the pivoted tongues can be turned in unison. During normal pump operation the tongues are held in line with the fixed vanes (i) ; but during starting or stopping the tongues are closed (ii). The regulating system thus resembles the gate mechanism of a

Francis turbine, but now its purpose is rather different: it is intended to control the potentially dangerous pressure-fluctuations that may arise when the pump is stopped, § 350.

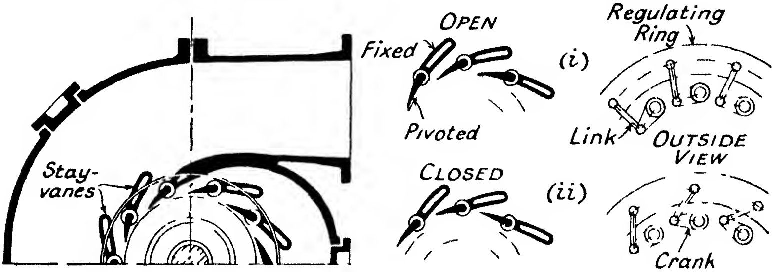


FIG. 105.– Pump with adjustable guide blades.

If the hydraulic conditions demand a two-stage construction, the disposition shown in Fig. 106 has been found satisfactory. The first stage consists of a pair of opposed single-inlet impellers working in parallel, each with its own suction branch, diffuser

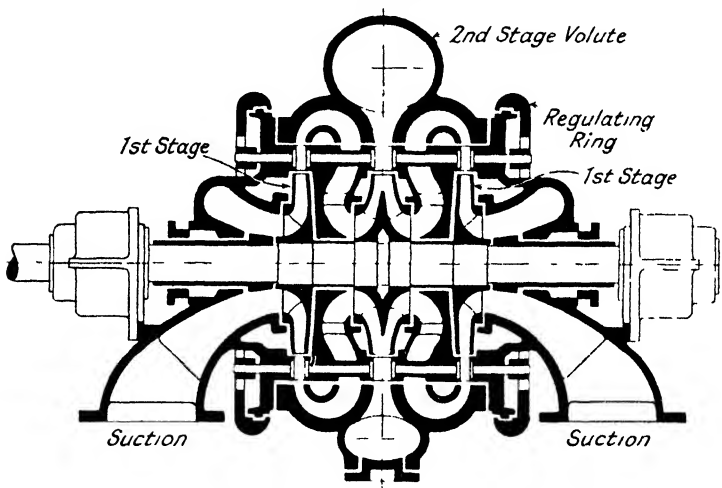


FIG. 106.—Two stage pump for hydraulic storage installation.

and return passages, § 122; the second stage has a double-inlet impeller and a volute. Pivoted guide-vanes are provided as in Fig. 105. The extremely heavy design of the whole unit is specially to be noted (*).

PART C

PERFORMANCE

CHAPTER XI

DEFINITIONS AND TERMINOLOGY

158. Aspects of Pump Performance. After a pump has been built and when it is ready for service, the designer naturally wishes to know how it behaves. By observations made under running conditions, he must satisfy himself that when driven at the specified speed, the pump really delivers the specified quantity of liquid against the specified head.

But there are wider aspects of performance than these. What will happen if the pump is compelled to work under a higher or a lower head than was originally foreseen? What will be the effect of varying the running speed? If the liquid supplied to the pump is changed, will the performance also be altered? For how long will the machine keep up its test-bed performance? Will it become unserviceable after two years or after twenty years? The answers to these questions may often be linked with the manner of installing the pump. We shall probably find that it is essential to study the installation as a whole, rather than to keep our eyes on the pump alone. Alterations in the length and diameter of the piping, or in the position of the pump in relation to the piping, may have a profound influence on the overall performance. Nor can we afford to neglect the very complex conditions that arise when the pumping set is being started up from rest, and when power is cut off from the motive unit.

If the purchaser and user may find it worth while to be informed on such matters, it is quite essential that the pump designer and the research engineer should so instruct themselves.

Only by patiently recording the widest range of facts about the biggest variety of pumps can the manufacturer make trustworthy forecasts about the performance of projected new machines. This array of facts must ultimately include many more than mere records of observations of speed, head and discharge; experimental methods must be devised to explore systematically the state of affairs within the pump.

159. Nomenclature of Performance Conditions. In this book the following terms will denote conditions of pump performance :—

Design Conditions. The pump is running at its normal designed speed, it is delivering its normal rated flow, and it is therefore presumably generating the stipulated head. The machine is then said to be operating at its *design point.*

Reduced-flow and Increased-flow. The designed speed remains unaltered, but the discharge is less than normal (reduced-flow, or part-flow), or greater than normal (increased flow). These variations in discharge are almost invariably accompanied by or caused by changes in the head generated. Reduced-flow conditions include the special case of zero discharge : the pump is delivering no liquid at all. Since this result is usually achieved by closing a throttle-valve on the delivery pipe, the term *closed-throttle* conditions is used synonymously with *zero-discharge* conditions.

(*Note*.—The terms *part load* and *overload* are sometimes used in place of reduced-flow and increased flow. But in this book they are rejected because part-load conditions in this sense, viz., reduced flow, may throw an *overload* on the driving motor, § 213.)

Universal Flow Conditions. The pump may run at any speed, either lower than the designed speed (reduced-speed), or higher than the designed speed (increased-speed). The flow may be either normal (designed) or reduced or increased or zero. The liquid may be water or any other fluid for which the pump is suited.

Geometrically-similar Conditions. Here we are concerned not with the actual pump itself, but with a reproduction of the machine on a larger or smaller scale. The reproduction may be a real or an imaginary one, but in any event the proportions of the original or prototype are assumed to be faithfully copied. The term geometrically-similar conditions, although perhaps

lacking in precision, will serve to indicate that what is in question is the performance of a geometrically similar pump. These conditions also can be studied under the general heading of Universal Flow.

Installed Conditions. Under this head we may conveniently group such departures from test-bed conditions as cannot readily be reproduced in the test plant. They may comprise :—

(i) Effect of piping layout, etc.
(ii) Effect of suction lift.
(iii) Effect of length of service, e.g. deterioration due to wear, corrosion, etc.

Transitory Conditions. This term will be used to describe the special type of installed conditions that are imposed on the pumping installation during changes of regime of any kind. These transient conditions occur during or immediately following the short periods of time in which the speed, the head, or the discharge are in course of variation. In general, the expression will apply particularly to the starting and stopping of the pumping set.

160. Assessing the Performance. Keeping steadily in mind the basic purpose of the pump, viz. to transmit energy to the liquid flowing through it, then the success of the machine when it actually begins to work can only be assessed by accurately measuring the energy increment received by the liquid. We must measure (i) the total energy of the liquid at a point or plane just before the liquid reaches the pump, (ii) the total energy just after the liquid leaves the pump ; and the difference between the two values will provide the desired information. Now *total energy* is defined as the sum of the position energy or geodetic head, the pressure energy and the velocity energy. If we were concerned only with conditions at a point, this summation might be easy. But in fact we are usually thinking about a plane—or two planes ; a transverse plane upstream of the pump and another one downstream. These may be coincident respectively with the suction and delivery flanges of the pump casing. It is possible but by no means inevitable that the pressure-head may be uniform at all points in a selected plane ; but it is in the highest degree unlikely that the velocity head will be similarly uniform. In fact we can say positively that the velocity distribution will not be exactly uniform

across the suction plane, and that it will frequently be still less uniform across the delivery plane. Apparently, then, we ought to make a number of velocity and pressure measurements at a number of positions in each plane, and thus compute a mean value of total energy at inlet and outlet. We *ought* to do this, but we hardly ever do it. The labour involved would be too serious. Instead, we use a single mean value of velocity energy based on the mean velocity at the pump flanges. It follows that pump tests as normally conducted may *never tell us precisely* what energy the liquid has received. There may always be an element of uncertainty concerning the velocity distribution, especially at the delivery flange. As a rule the range of uncertainty falls within the limits of error permissible in pump tests, but that does not absolve the testing staff from the need for special care when making measurements at the pump outlet.

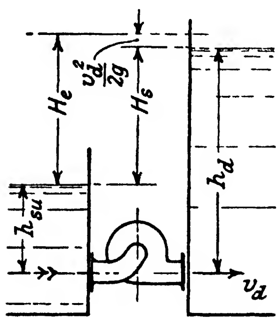


FIG. 107.—Simple pumping circuit.

161. Dead Head and Total Head. The simple layout sketched in Fig. 107 seems to offer opportunities for easily assessing energy changes. The pump suction flange is directly bolted to an inlet chamber, and the delivery flange is similarly connected to an outlet tank; at the delivery flange the mean velocity is v_d. In the two tanks the surface levels are assumed to remain unaltered. At the pump inlet, the total energy of the liquid will be represented by the positive head h_{su} above the pipe axis; at outlet, the liquid has a pressure energy h_d and a mean velocity energy $\frac{v_d^2}{2g}$. If $H_s = (h_d - h_{su})$ is the difference of surface level—the *static* or *dead* head—then evidently the nominal energy increment is $H_s + v_d^2/2g$.

Now let the two chambers be spaced some distance apart, with horizontal lengths of uniform pipe interpolated, Fig. 108. If the pump speed and discharge have remained unchanged, this alteration to the system cannot disturb the internal working of the pump: the energy increment H_e must still have its original value. Yet the dead head, or vertical height through

which the liquid is lifted, has fallen from the value H_s to the value H_s'. The reason is clear : some of the energy the pump has given to the liquid is quickly dissipated in pipe friction. The conventional method of plotting pressure-head and total energy enables the changes in the energy-content of the liquid to be readily traced as the liquid passes through the system.

Actual pumping installations are usually a good deal more complicated than this ; they include foot-valves, bends, elbows, etc., and there may also be variations in pipe diameter. More

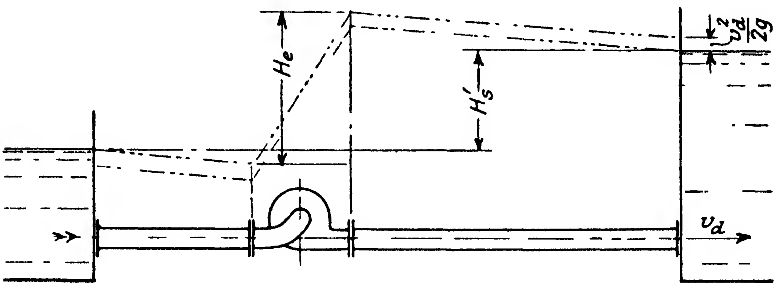


FIG. 108. Circuit including static head and friction head.

comprehensive methods of assessing effective head are therefore essential.

162. Manometric Head and Effective Head. In the generalised system depicted in Fig. 109, energy *losses* can be grouped in the following categories :—

(i) Loss at entry, e.g. in the foot-valve and strainer, § 290, h_{in}.

(ii) Pipe friction losses, = h_f.

(iii) Eddy losses due to bends, elbows, partially-closed valves, etc. h_l.

(iv) Velocity energy dissipated in the outlet tank, $= \dfrac{v_p^2}{2g}$

(where v_p = exit velocity from delivery pipe).

(v) Losses in the suction system only, excluding inlet loss, = h_{fls}.

Now the net or ultimate energy gain experienced by the liquid, from the moment it begins to move in the lower tank to the moment it finally comes to rest in the upper tank, is equivalent to the dead head or static lift H_s. Evidently, therefore,

the effective energy increment that must be imparted by the pump must be the sum of the dead head and all the energy losses. The summation is not at all an easy one—in fact unusually good judgment is required in estimating energy losses in such conditions. Fortunately there is no need to rely upon so uncertain a method. The pressure-difference generated by the pump can be measured with sufficient accuracy by means of pressure-gauges connected to tappings at the pump flanges, and

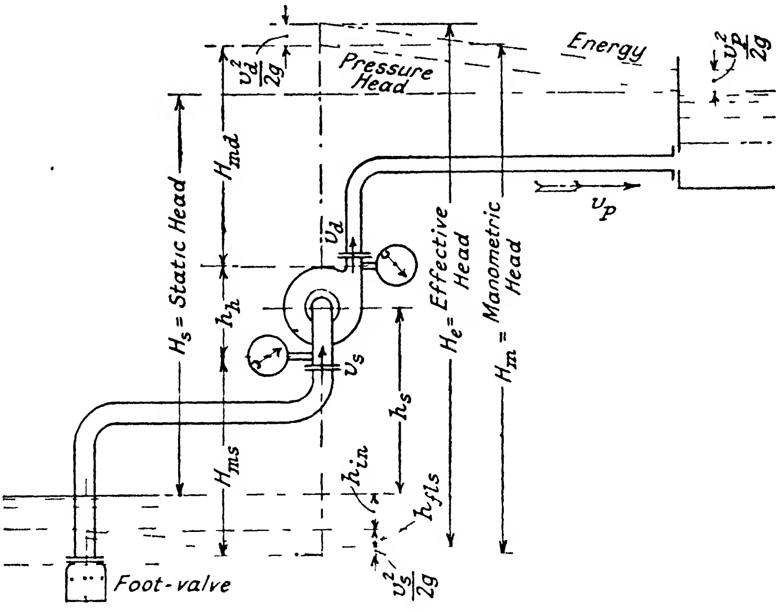


Fig 109 Elements of generalised pumping system

when these readings are corrected for velocity head the result yields the desired value of the energy increment with as small an error as we can reasonably hope for.

If H_{ms} = reading of suction gauge } both in terms of *head*
H_{md} = reading of delivery gauge } *of pumped liquid.*
v_s = mean velocity at suction flange,
v_d = mean velocity at delivery flange,
h_h = vertical height or difference in level between gauges,

then the *effective head* on the pump, H_e, is represented by

$$H_e = H_{md} + \frac{v_d^2}{2g} + H_{ms} - \frac{v_s^2}{2g} + h_h.$$

The *manometric head* on the pump, so called because it depends on manometer or pressure-gauge readings, is defined as :—

$$H_m = H_{ms} + H_{md} + h_h.$$

163. Standardised Estimation of Head. To bring the above expressions within standardised conditions, we may require the gauges to be set *at the level of the pump horizontal axis*, thus making h_h = zero ; or alternatively the actual readings may be adjusted to give the same effect. On this understanding, the various meanings to be attached to the term " head " can finally be collected together thus :—

Total static head or *dead head* (Difference in surface level between suction and delivery chambers) $= H_s$

Static suction head or *suction lift* (Vertical distance between pump axis and surface level in suction chamber) h_s

Manometric head (Increase of pressure head generated by pump) $= H_m$

$$= H_{ms} + H_{md}$$

Manometric suction head (Standardised reading of suction gauge) H_{ms}

$$h_s + h_{in} + h_{fls} + \frac{v_s^2}{2g} \quad . \quad \textbf{(11-1)}$$

Effective head (Increase of total energy generated by pump) $= H_e$

$$= H_{md} + \frac{v_d^2}{2g} + H_{ms} - \frac{v_s^2}{2g} \quad . \quad \textbf{(11-2)}$$

$$= H_s + h_{in} + h_f + h_l + \frac{v_p^2}{2g} \quad . \quad \textbf{(11-3)}$$

Henceforward the standardised terms just defined will invariably be used. **(Example 18)**

(*Note*.—American engineers sometimes use the term *dynamic head* in place of *effective head*.)

Naturally these expressions must be used with discretion. Should the pump have a *positive* inlet head, for example (Figs. 107 and 108), so that there is a positive pressure at the suction flange, then the pressure increase across the pump flanges will

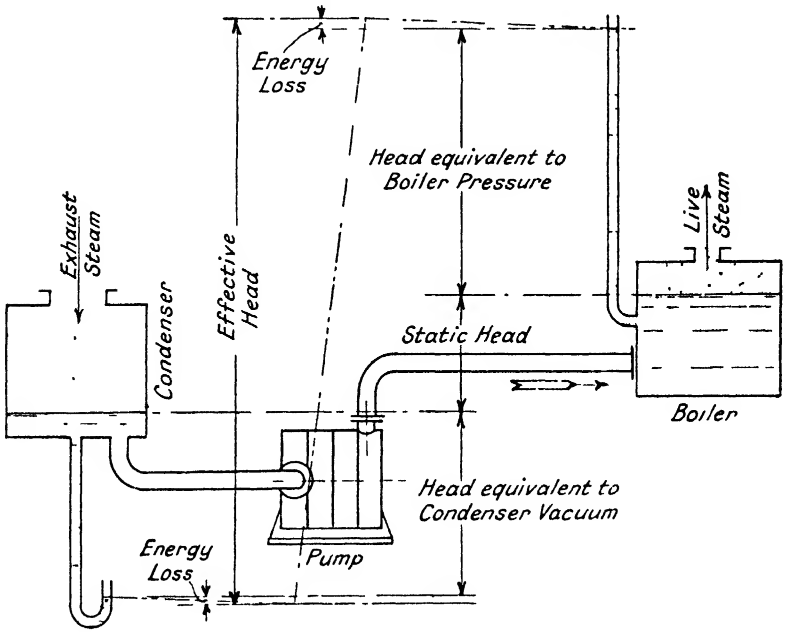


FIG. 110. Elements of boiler-feed circuit.

be proportional to $H_{md} - H_{ms}$, and not to $H_{md} + H_{ms}$. Further, equations (11-1) and (11-3) are only valid if the liquid in the suction and delivery chambers is freely exposed to the atmosphere. An installation where this stipulation is by no means fulfilled is shown in Fig. 110; a pump (or in practice a group of pumps) draws water from a condenser and delivers it into a steam boiler. Since the velocity heads are here extremely small in relation to the pressure heads, they are disregarded in the diagram. We observe that the effective head on the pump is now the sum of (1) the static head, (2) the

energy losses in the pipe system, (3) the head equivalent to the condenser vacuum and (4) the head corresponding to the boiler (gauge) pressure. If we plot *what would be* the position of the levels of the free surfaces if these were allowed to develop, Fig. 110, then equation (11-3) can be applied without further correction. **(Example 19)**

In assessing the value of the effective head under any circumstances, it is almost invariably helpful to plot or at least sketch the hydraulic gradient and energy lines, as in Figs. 108 to 110. Nor must it be forgotten that the pump has no means of discriminating between the different ways in which the total head may be imposed on it. It is a faithful but not very intelligent servant, which under stated conditions will generate exactly the same head no matter whether the head is created by gravity or by friction or by throttling or by artificial positive or negative pressures, or by any combination of these.

164. Energy and Hydraulic Efficiency. What are the mathematical terms that have been used for expressing effective head or energy increment? Nominally the terms are units of length— feet or metres —but basically the terms are *energy per unit weight of liquid*, e.g., foot-pounds per pound of liquid or kilogram-metres per kilogram of liquid. To say, then, that a pump generates an effective head of 28 ft. is a convenient abbreviation of the statement: the pump is giving to each pound of liquid an energy increment of 28 ft./lb. Here is a clue to finding methods of expressing the efficiency of the pump or of the pumping installation. Since the term efficiency as used by engineers always has the sense: $\dfrac{\text{Output}}{\text{Input}}$: since it is a ratio or dimensionless quantity: then if effective head is taken as a measure of output, it follows that energy input must also be expressed in terms of energy per unit weight. According to the fundamental theory of rotodynamic pumps, § 11, this input is represented by $E = \dfrac{V_2 v_2}{g}$, where V_2 is the tangential or whirl velocity component impressed on the liquid, and v_2 is the rim velocity of the rotor. On this basis the ideal efficiency of the ideal impeller was seen to be $\dfrac{gH}{V_2 v_2}$. . (§ 13)

If for the ideal head we substitute the observed effective head H_e, and if we replace the ideal whirl component V_2 by the true whirl component V_n, § 24, then the actual pump efficiency may be written :—

$$\eta_h = \frac{gH_e}{V_n v_2}.$$

This value η_h is known as the *hydraulic efficiency.*

Because of the experimental difficulties in estimating the numerical value of the whirl component, § 17, precise values of the hydraulic efficiency η_h can rarely be established. Nevertheless arbitrarily-assumed values, § 93, serve as an indispensable basis for pump design. The factors that influence these values are examined in §§ 193 to 195.

165. Power and Gross Efficiency. An alternative method of assessing the behaviour of rotodynamic pumps is to use the expression :-

$$\textit{Gross or overall efficiency} = \eta_m = \frac{\text{Power output}}{\text{Power input}}.$$

The various terms of this expression can be evaluated thus :—

Power input, or *Shaft horse-power.* Denoted by the symbols S.H.P., or P_s, this is the horse-power fed into the pump shaft. If the pump is direct-coupled to its motive-unit, then the S.H.P. is identical with the B.H.P. output of the driving engine or motor.

Power output, or *Water horse-power.* In general, horse-power is represented by

$$\frac{\text{Energy per second}}{K_p},$$

where K_p is the horse-power constant, or energy per second equivalent to 1 horse-power.

For the particular case of a rotodynamic pump, the energy per second imparted to the liquid is expressed by

Weight per second $\times$ energy per unit weight.

We can therefore write :

$$\text{W.H.P.} = P_w = \frac{W H_e}{K_p} = \frac{Q w H_e}{K_p},$$

where P_w = water horse-power.

W — weight of liquid pumped per second.
Q — volume pumped per second.
H_e — effective head generated.
w — density of liquid passing through pump.

For routine computations it is convenient to use the form :—

$$P_w = \frac{\text{Discharge} \times \text{head or pressure}}{\text{Divisor K}}.$$

According to the units chosen, the value of the divisor K will be :—

Units of Discharge	Units of Effective Head or Pressure	Divisor K
Pounds per second	Feet head	550
Gallons of *cold water* per minute	Feet head	3,300
Cubic feet per second	Pounds per square inch	3·82
Kilograms per second	Metres head	75
Litres per second	Kilograms per square centimetre	7·5

Gross efficiency. The value of the gross efficiency,

$$\eta_m = P_w / P_s,$$

can now immediately be computed. Since the gross efficiency is easier to evaluate, and is more generally useful, than the hydraulic efficiency, the term *efficiency* henceforth used in this book will always mean *gross* efficiency unless otherwise stated. The complex influences that affect its numerical value are explained in Chapters XIII to XVI.

Chart III, facing page 480, will assist in giving approximate values of power, efficiency, etc., if the necessary data are available.

Static Efficiency. In a pumping installation designed to raise water through a dead head H_s, it will be seen from Figs. 108 to 110 that the actual energy H_e to be impressed on the liquid may depend materially upon the design of the inlet

and outlet passages. In other words, the useful W.H.P. output of the installation *as a whole*, as distinct from the pump alone, can be expressed by : $\frac{WH_s}{550}$ (in foot units). The corresponding value of the efficiency, viz. $\frac{(WH_s)/550}{\text{S.H.P.}}$, is sometimes known as the *static efficiency* of the installation.

CHAPTER XII

TESTING OF PUMPS

166. Types and Conditions of Tests. In general, rotodynamic pumps may require to be tested (*) in one or other of three sets of conditions, viz. :—

Routine works tests.

Acceptance tests, either at works or on site.

Special tests.

If the purchaser's representative is present at a works test, then the trial may also serve as an acceptance test. Pumps that are too big to be run at the works testing plant must necessarily be tested on site, after erection. In any event the observations required are those of (i) Speed, (ii) Suction and delivery head, (iii) Discharge, (iv) Power input. From the test figures, values of effective head, power output and gross efficiency can be computed, as defined in §§ 163 to 165. As in any other running test, the attendants or officials conducting the trial must verify that the machine runs smoothly, quietly, without overheating, and so on.

It will be convenient to enumerate first the various types of measuring appliances that may be found suitable, and then to examine layouts of complete testing equipments appropriate to different circumstances. Details of the construction and use of the measuring gear will be found in specialised publications (*).

167. Measurement of Speed. Although a revolution counter, in conjunction with a stop-watch, will usually give a

more reliable indication of rotational speed than a tachometer, yet a tachometer may often be essential in order to verify that the speed is held constant during the test run. If a tachometer only is used, it should be accurately calibrated. In electric tachometers, the indicating instrument is in effect a voltmeter actuated by the current generated by a small dynamo coupled to the pump shaft. They have the advantage that the indicating dial—which is graduated in terms of rotational speed—can be mounted on the instrument panel which serves for the entire pumping installation. The outfit needs special calibration if used for precise testing.

If the pumping set under test includes a synchronous motor taking its supply from a frequency-controlled network, then a very accurate knowledge of the shaft speed may be available.

168. Measurement of Head or Pressure. (i) *Direct Observation of the Free Water Surface.* The water level is read from fixed graduated scales mounted in the suction or delivery chambers. Naturally it is only possible to take both readings in this way if the static head is relatively low, or if there is accurate information about the vertical distance between the zeros of the scales.

(ii) *Open Water Columns.* Here the levels of the free liquid surfaces are read from graduated glass gauge tubes. If the glass gauge tubes are connected direct to the suction and delivery branches of the pump, free surfaces are unnecessary.

(iii) *Float-gauges.* The floats may ride either on the water surfaces in the suction and delivery chambers, or they may work in special vertical cylindrical gauge-wells communicating with the chambers.

(iv) *Mercury Columns.* When the heads to be measured exceed 10 ft. or so, water columns may be inconveniently high. Mercury manometers are suitable for heads up to 50 ft. or more, and they can be read with great accuracy.

(v) *Differential Mercury U-tube.* If one leg of the U-tube is connected to the pump suction branch, and the other leg to the delivery branch, then the gauge gives a direct indication of the manometric head on the pump, § 176 (ii).

(vi) *Spring-loaded Gauges.* This general term covers such instruments as Bourdon, diaphragm, and bellows gauges, in which the applied pressure causes an elastic metallic element to

yield, the deflection being magnified and indicated by a pointer moving over a scale graduated in terms of head or pressure. They are the commonest and often the most acceptable devices for pressure measurement—either positive or negative; but unless they are carefully handled and frequently calibrated, their indications are apt to be unreliable.

(vii) *Power-operated Pressure-measuring Machines.* These instruments have recently been developed for the most precise indication of high pressures—say from a few hundred to several thousand pounds per square inch. Based on the principle of the dead-weight gauge tester, they are elaborate and costly in construction, and are likely to be chosen only when pumps of the highest efficiency are to be tested in research institutes (*).

169. Measurement of Discharge. (i) *Absolute Methods.* Here the liquid is collected in measuring tanks or receptacles of known capacity, during a measured period of time. The quantity collected during the run is found either by weighing the tank and contents, or by observing the change in surface level. As no other measurements are involved other than those of time and either length or weight, absolute methods are the most reliable of all types of discharge gaugings. They should invariably be used, if by any means possible, for controlling the results of indirect gauging methods.

(ii) *Free-flow Methods.* The liquid discharged from the pump flows freely and continuously through a gauging orifice or over a gauging weir. The rate of flow is computed from observations of the head over the orifice or weir. A number of orifices may work in parallel, and two or more weirs may work side by side. For small discharges, the triangular weir is preferable; for larger flows, the suppressed rectangular weir will serve.

(iii) *Meters in Closed Pipes.* (*a*) "Quantity" or *total-flow* meters when interposed in the pump delivery pipe indicate the total weight or volume of liquid pumped, during the interval between two successive readings of the dials. By dividing this quantity by the measured time interval, the desired mean rate of discharge is found. The helical type of inferential meter, suitable for pipes from about 3- to 12-in. diameter, may give sufficiently accurate results for routine tests, especially if regularly calibrated.

(*b*) *Rate-of-flow* meters indicate the instantaneous discharge, without the need for observations of time. What is actually observed is the differential head generated as the liquid flows through a constriction in the pipe. The metering outfit thus comprises two main elements, (i) the primary element or constriction which is actually interpolated in the pump delivery pipe, and (ii) the differential-head gauge which indicates pressure differences. Types of constriction or primary element usually chosen are the Venturi tube, the orifice, and the flow-nozzle.

(iv) *Special Methods.* When very large pumps must be tested on site, a special technique may be required for gauging the discharge. Multiple current-meters, either in the delivery pipe or in an open delivery outlet channel, are sometimes suitable ; alternatively, the Pitot-tube or the Pitot-sphere may be preferred.

170. Measurement of Power Input. (i) *Torsion Dynamometer.* This instrument is particularly suitable for the conditions of high speed and uniform torque that characterise most rotodynamic pumps of small and medium size. It is interposed axially between the driving motor and the pump shaft. The actual torque is transmitted from the one shaft to the other through a rod of tempered steel that is so thin that it twists appreciably ; the angle of twist is directly observed by an optical device, and the corresponding value of the torque is read from a calibration chart. As the shaft speed has meantime been observed, the power input to the pump can at once be computed. The diameter of the elastic steel transmission element is chosen to suit the expected value of the torque, and as the calibration curve for each of the elements can be checked by a dead-weight method, the overall error can be kept well within ± 1 per cent.

(ii) *Swinging-yoke Motor.* Here the electric driving motor is direct-coupled to the pump shaft, but the motor is of special construction, having its yoke or frame mounted on ball bearings co-axial with the shaft bearings. As soon as the motor is energised, the frame tends to revolve in a direction contrary to that of the armature or rotor ; and this tendency is neutralised by a jockey weight that can slide along a horizontal arm projecting from the frame. When the set is running steadily, the weight is adjusted by hand until the torque-arm takes up its zero position. From the known mass of the jockey-weight,

and its position on the arm, the torque can be directly worked out, and, in turn, the power transmitted is found. Limit-stops, and possibly an adjustable dashpot, control the movements of the motor frame. Here again is a method which is subject only to errors that can either be measured or closely estimated.

(iii) *Inferential Methods.* The power transmitted from an electric motor to the pump it is driving can be estimated from the instruments that measure the electric supply to the motor. The estimate will be accepted with greater confidence if a brake test of the motor has been made, so that known voltmeter and ammeter readings correspond to a measured B.H.P. motor output. For very large motors that cannot be tested in this way, the calculations of the motor designers must be used to interpret the electrical readings.

If a steam-turbine or an oil-engine drives the pump, evidently it will not be easy to assess the true H.P. fed to the pump shaft.

Planning the Works Testing Plant

171. Guiding Considerations. The essentials of the plant are the pumps themselves and the measuring apparatus; but before the tests can be carried out, a supply of liquid and a supply of power must be provided. If the liquid is cold water at atmospheric temperature, there would seem to be little difficulty here. Yet as the size of the pump increases, so also does the problem increase of contriving large enough reservoirs that are not prohibitively costly. On the other hand, attempts to restrict the size of the pipes and tanks may falsify the test results by imposing unfavourable inlet conditions on the pump. If the pump is to be tested with hot water or hot oil, then it is still less practicable to circulate very large quantities of liquid.

The arrangements for power supply may have to be quite comprehensive. If the pump only is to be tested, then one or other of the test-bed electric motors will serve. But the customer may require a running test of the complete pumping set, and since the motive unit may be an electric motor, an oil engine, or a steam turbine, therefore electric current of various types should be on tap, and there should be a high-pressure steam boiler.

The cost of running the tests can by no means be neglected. Centrifugal pumps of small and medium size are built for sale in so highly competitive a market that unnecessary expense in testing is just as inadmissible as it is in machining, assembling or despatching the pump. With larger units, the cost of the power absorbed in protracted tests may be sufficiently serious as to encourage other means of collecting the desired information, § 183.

Fortunately there is little difficulty in simulating on the test-bed the conditions of working pressure the pump will have to meet. A partially-closed throttle-valve in the delivery pipe will generate any stipulated head-drop; nor is the pump able in the slightest degree to discriminate between such an artificial head and the dead head that may be imposed upon it after installation. Although a standard sluice-valve usually suffices, it is worth remembering that in tests of high-capacity, high-pressure pumps, energy is dissipated in the valve at the rate of several hundred horsepower. Throttle-valves of special design may here be necessary.

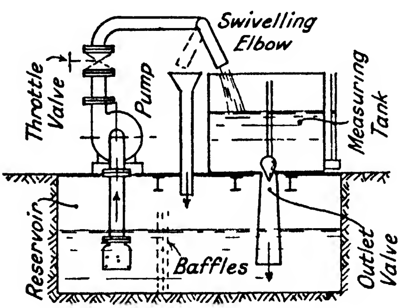


FIG. 111.—Simple test rig with measuring tank.

On the inlet side of the pump, throttling is less satisfactory as a means of creating an artificial suction head, but it is not difficult to contrive alternative methods, § 175.

172. Open-circuit System with Measuring-tank. The term "open-circuit" here implies that the liquid both on the suction and delivery side is contained in chambers or reservoirs having a free surface exposed to the atmosphere. The suction container may be a large underground tank serving a number of pumps simultaneously. Fig. 111 indicates the simplest possible layout. The pump, bolted to the test-plate which forms the cover of the underground reservoir, draws water through a short suction pipe, and discharges it through the throttle-valve and so to a swivelling elbow which directs the flow either into the measuring tank or to waste. The quantity of water in the

tank is shown by a glass gauge-tube. A conical outlet-pipe or draft-tube, provided with a stream-lined valve, permits the tank to be quickly emptied; baffles ensure that the pump draws from reasonably still water. A tank mounted on a weighing machine would be just as satisfactory as the volumetric tank shown in the sketch.

An error that may or may not be negligible results from the fall in level in the suction reservoir as the measuring tank fills. Undoubtedly if the pump speed and the throttle-valve setting are held steady during the run, as they should be, the variation in suction lift will alter the total effective head on the pump, and the discharge will be correspondingly affected. In the case of (*a*) an underground tank of very large surface area, (*b*) a high delivery head, say of several hundreds of feet, then it is possible that the alteration in suction level may legitimately be disregarded. But it could certainly not be disregarded if there were a 3-ft. drop in level and a total head of (say) only 30 ft. One way out of the difficulty is this: instantaneous readings of suction and delivery head are taken exactly half-way through the run, i e., at the middle point of the period in which the measuring tank is being filled. The assumption would then be that the pump discharge, which in fact is continuously diminishing throughout the test run, attains its mean value half-way through the run. It is by no means certain that such an assumption would be justified.

Another method, which now really ensures uniformity of discharge, is to station an attendant at the throttle valve, with orders to regulate it so as to keep the observed manometric head constant. In other words, as the suction head increases, the delivery head is reduced correspondingly. Probably the back-lash inevitable in a normal sluice-valve would make the regulation rather difficult; but if the adjustment were made by using a small needle- or globe-valve set in a by-pass circuit, in parallel with the main throttle-valve, then the desired sensitivity should be attainable.

173. Other Measuring-tank Systems. If the suction level could artificially be held steady while the measuring-tank were being filled, then all the uncertainties just pointed out could be swept away. A method of doing so is proposed in Fig. 112 (i). Here the pump and the test tanks form a

self-contained system, set above ground level. There are twin measuring tanks, separated by a smaller suction tank. Twin valves mounted on a horizontal spindle and worked by an external lever direct water from either of the main tanks into the central compartment, from which it flows into the suction pipe and so to the pump. A swivelling elbow or the like fitting switches the water from the delivery pipe into the appropriate measuring tank. The flow into the central or suction compartment is controlled by a float-operated butterfly valve, so disposed that the water level in this compartment never varies by

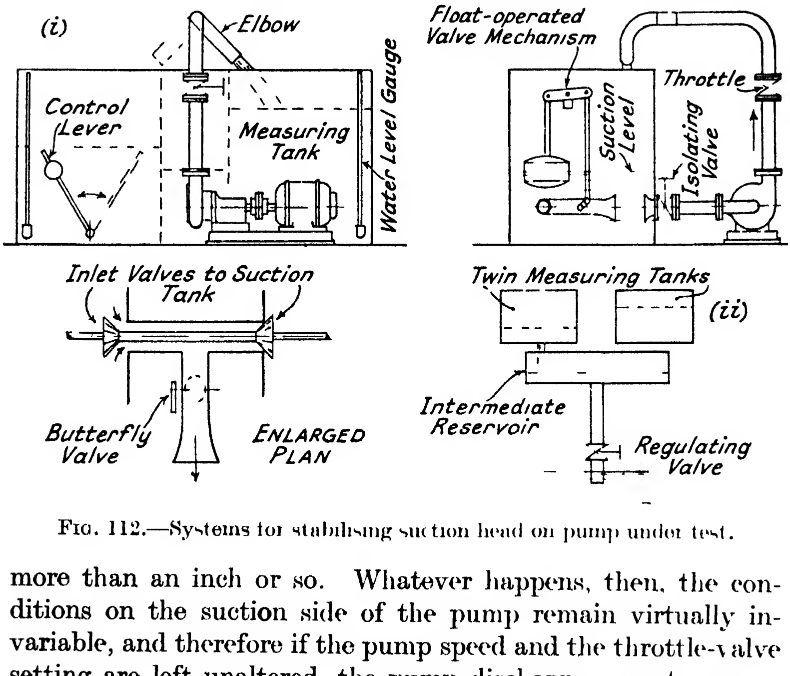


FIG. 112.—Systems for stabilising suction head on pump under test.

more than an inch or so. Whatever happens, then, the conditions on the suction side of the pump remain virtually invariable, and therefore if the pump speed and the throttle-valve setting are left unaltered, the pump discharge cannot vary.

As the system has been found to work very well with water, it may also be recommended for small amounts of other liquids, e.g. oil. The inset diagram, Fig. 112 (ii), shows how almost the same results might be attained when the pump draws from a communal underground reservoir. The twin measuring tanks are mounted as high as possible above the lower tank, and they are emptied into an intermediate receptacle from which a drain pipe returns the water to the lower reservoir. A throttle-valve

in the drain-pipe is adjusted so that the level in the intermediate tank fluctuates between the full and empty marks. Then so long as the pump discharge keeps steady, the return flow back into the lower reservoir will vary through so small a range as almost to damp out changes in the suction head on the pump. The greater the surface area of the intermediate tank, the less will be the fluctuation of level.

Any system of twin calibrated tanks—Fig. 112 (i), (ii)—has this weighty recommendation: it permits continuous flow measurement to be carried on. While one of the tanks is being filled by the pump, its fellow is being drained.

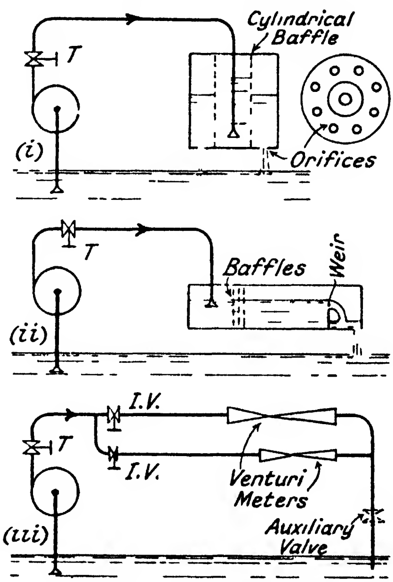


FIG. 113 Use of orifices, weirs, and meters for flow measurement. (Note T throttle valve, I.V. isolating valve.)

174. Flow Measurement with Orifices, Weirs, and Meters. Measuring-tanks undoubtedly provide the most reliable method of gauging the discharge of pumps. But the apparatus is cumbersome, the tests take a relatively long time to carry through, and there may be practical complications of the sort just described. For many purposes the desired information can be found with sufficient accuracy by the use of orifices, weirs, or meters, § 169. The diagrams in Fig. 113 show typical layouts. Special consideration must often be given to the question of gauging small flows. Both with the orifice, diagram (i), and the constriction meter, (iii), the head to be measured (which is of course entirely distinct from the head *generated by the pump on test*) varies as the square of the discharge, with the result that the head corresponding to 10 per cent. discharge is only 1 per cent of the head corresponding to full discharge. It is impossible to measure accurately so small a head. The difficulty can be evaded by using multiple measuring

devices. Thus the circular orifice tank, Fig. 113 (i), has in its base a number of gauging orifices, of which only one would be used for small flows—the others being suitably blanked off —while all the orifices would be opened for maximum flow. In Fig. 113 (iii), a large and a small Venturi meter are set in parallel, and they can be used singly or in combination. Although weir characteristics are more favourable in this respect, it may be nevertheless convenient to use, for example, a rectangular and a triangular weir in parallel.

The constriction meter, diagram (iii), has a unique advantage : it responds nearly instantly to changes in discharge, and thus there is no need to wait after any adjustment before taking a reading. If it is desired to run the pump at a stipulated discharge, the throttle-valve can be manipulated to bring the indicating column straight away to the corresponding graduation. This rapidity of response permits the points on a characteristic curve, § 212, to be established within the space of a few minutes only. On the other hand, the orifice tank or the gauging weir, Figs. 113 (i) and (ii) may be subject to a considerable time lag. The only way of making sure that the water levels have taken up their new positions after changes of flow is to take two readings in succession and verify that they are identical. Moreover, variations in the level of the main underground reservoir—such as might occur when this reservoir serves a number of pumps simultaneously—will affect the head generated by the pump when systems (i) and (ii) are used, possibly necessitating slight regulation of the throttle valve, § 172 ; but system (iii) will remain unaffected. The additional valve seen in the outlet pipe in diagram (iii) can be regulated, if desired, to ensure that nowhere in the metering system is there a negative head.

175. Closed-circuit Systems. These may be necessary when the test conditions enforce the use of the minimum quantity of liquid relative to the size of the pump. Such conditions evidently arise when the pump approaches the limiting size that can possibly be given a works test at all, or when the liquid to be handled is hot water or oil or other non-aqueous substance. The liquid is now circulated through a closed system having only one container, and probably the free surface in this vessel may not be exposed to the atmosphere.

A good example of a test rig (*) for a large low-lift pump of 1 cu. m./sec. capacity (13,000 gall./min.) is illustrated in Fig. 114. It embodies a sharp-edged measuring orifice, and straightening-vanes for rectifying the flow before the water reaches the orifice. The throttle-valve is necessarily set on the downstream side

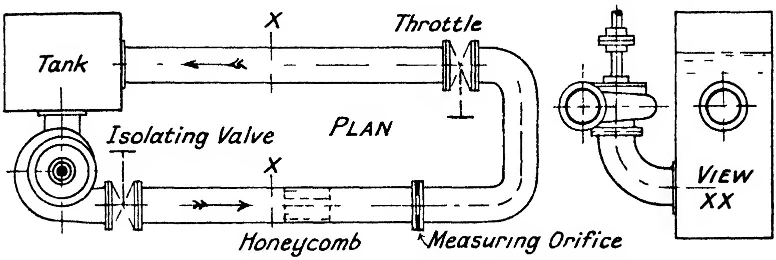


FIG. 114. Closed circuit test rig for low head pump

of the constriction. For measuring the power input a torsion dynamometer was used, § 170.

Fig 115 (i) suggests a test circuit for a multi-stage boiler-feed pump. The positive inlet head under which such pumps often have to operate can be simulated by forcing air into the closed drum ; and the desired water temperature is attained by injecting steam into the liquid. As an unvarying weight of liquid continuously circulates, the changes in the delivery head

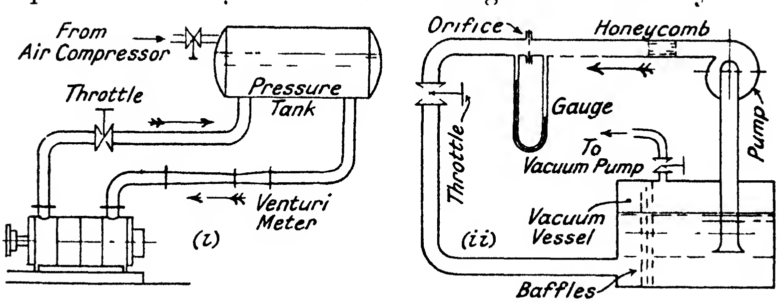


FIG. 115. Closed circuit systems for use with (i) high delivery heads, (ii) high suction heads.

induced by throttling have no effect on the suction head. By draining off the water and replacing it by oil of the stipulated consistency, the test rig becomes suitable for oil pumps.

Pumps designed for specially high suction lifts should preferably have a shop test under those identical conditions. As a throttle-valve on the suction pipe itself would unfairly disturb

the inlet conditions, and as the suction head so established would be at the mercy of every alteration of discharge, the receptacle of the closed-circuit system might now be set on the suction side of the pump, Fig. 115 (ii). An air exhausting pump connected to the drum could maintain the pre-determined vacuum, which here also would be quite unaffected by manipulation of the main delivery throttle-valve. To remove the danger of air leaks into the suction side of the system, the drum and adjoining pipe-work might be entirely immersed in a larger reservoir of water.

In all closed-circuit systems, most of the energy-input to the pump is ultimately dissipated in heating up the liquid. To prevent excessive rise of temperature, therefore, cooling coils or the like may be essential.

176. Some Practical Details. (i) *Location of Pressure-tappings.* As there must inevitably be some amount of uncertainty in assessing the total energy received by the liquid in passing through the pump, § 160, it is manifestly desirable to limit this chance of error in all possible ways. Good judgment in choosing the points of pressure-measurement is one of the best methods of insurance. Another is to give the liquid a fair chance at inlet and outlet ; straight lengths of pipe should be interposed here, unless there are unfavourable conditions on the site which must be reproduced. Examples of bad and good practice are illustrated in Fig 116 (i) and (ii). Tappings located on pipe-bends inevitably falsify the gauge readings. If bosses are cast on the suction and delivery flanges themselves, then these can be tapped directly for the small pipes leading to the gauges. It is especially in low-lift pumps of high specific speed that flow conditions across the outlet flange may vary widely (*), and the variation may be more than ever acute at reduced rates of discharge, § 207. Multiple tappings equally spaced around the periphery of the outlet flange, forming a circumferential manifold pipe, may then be advisable, Fig. 116 (iii).

(ii) *Location of Pressure Gauges.* Fig. 109, § 162, clearly showed how the estimation of the effective head depends upon the position of the gauges themselves. If only the total manometric head H_m is to be measured on the test-bed, and if its value is not too great (say less than 50 ft.), then it can be read directly by a U-tube mercury gauge, § 168, connected across the

pump, Fig. 116 (iv). In this event the position of the gauge is immaterial. This particular diagram may remind the electrical engineer of the way he measures electrical pressure-differences by a voltmeter coupled across the terminals of his machine, while his own ammeter set in series corresponds to the hydraulic flow-meter.

When individual single-column mercury gauges are specified for measuring the suction and delivery heads, it may be worth while mounting them so that the mercury level in the containers can be brought exactly into the plane of the pump axis, Fig. 116 (v). Then the readings on suitable scales will give

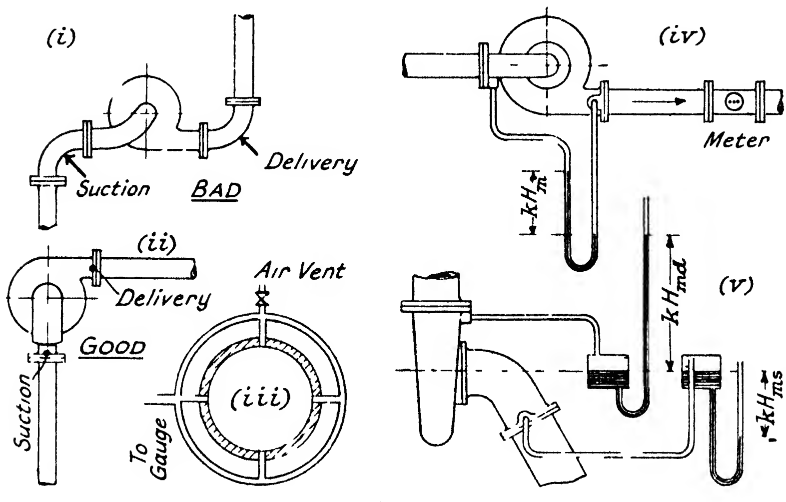


FIG. 116. Details of pressure tappings, etc.

at once the desired standardised suction or delivery manometric head, H_{ms} and H_{md}, relative to the pump datum plane, § 163.

(iii) *Baffles, Air Exclusion, etc.* As the pump and pipes grow bigger in relation to the size of the tanks and reservoirs, it becomes correspondingly important to baffle the flow at the appropriate points. On the suction side of the circuit, judiciously placed grids of wood or screens of perforated metal will damp out major eddies before the liquid actually reaches the inlet pipe ; on the delivery side, honeycombs or straightening vanes, Figs. 114, 115, will give the constriction flow-meter a fair chance of doing its work properly.

Test-bed operators of all people do not need to be told about the troubles that air bubbles in a pump circuit can induce. If, therefore, a test rig involves a stream of liquid splashing freely from the delivery pipe back into the underground suction reservoir, they will take very good care to prevent the resulting bubbles from drifting across into the pump suction pipe. It may happen that the only effective safeguard is to suppress the splashing by carrying the return pipe below the level of the suction reservoir, Fig. 113 (iii).

Other possible points of ingress for air bubbles are the vortices that may form around the mouth of the suction pipe. The remedy is clear: the pipe entry must be deeply submerged.

177. Organisation of a Test Plant. The first thing the superintendent of a projected test plant will want to know is the size and output of the pumps to be tested. Small and medium sizes can all be given a routine test, but it will be well to keep an eye on the possibility of stretching the accommodation in an emergency to receive a big pump as well. The foundation of the plant is the most capacious possible measuring basin that the allotted space or the allotted funds will allow. If it is made completely water-tight, if necessary even by providing a continuous unbroken welded sheet-steel lining; and if all inlet and outlet pipes can on occasion be completely disconnected; then tedious and uncertain leakage corrections can be eliminated. Furthermore, if the walls are built square and plumb, the tank capacity can be established by direct measurement. As all drain openings are forbidden, the tank must be emptied by a service pump drawing from a sump: a single-stage borehole type pump is often suitable. Normally the basin may serve as the common suction tank for routine tests; on special occasions routine testing is suspended and the basin is used either for calibrating the test-bed flow-meters or gauging weirs, or else as a measuring tank for testing outsize pumps.

Pumps that fall within the ordinary capacity of the testing equipment will be of two sorts: either they will be machines in current production intended directly for sale or stock, or they will be new or experimental designs. The first category, as already suggested in § 171, must be studied in the same frame

of mind as the machine-shop superintendent's when he checks the floor to floor time of the pump casing on the boring-mill. It may even be worth while to contrive quick-release clamps and fixtures for mounting the pump in the test rig. On the other hand it would be poor economy to hustle a tentative new design of pump prematurely off the test-bed. This applies especially to the scale-model pumps built solely to control the design of the actual pump which will itself be far outside the test plant's capacity, § 184. Patient testing and judicious modification may here prove a highly profitable investment.

When gauging-weirs or gauging-orifices are preferred, § 169, a question to be settled is this : is it better to use fixed gauging tanks, which may involve concrete channels or the like, and to bring the pump to the tank ; or would it be more convenient to make portable sheet-metal tanks which the crane can set down alongside the pump ? It is true that after each transfer the portable gauging tank may have to be accurately levelled, but on the other hand the tank can very easily be brought up to the main calibration basin. As for the gauging tackle for pressure and head, portability is never a disadvantage here. Mercury or water columns especially can be set up against a stoutly-built stand, some of the gauge tubes serving for manometric head and others perhaps for the secondary element of the constriction type of flow-meter. If the stand is brought near to the throttle-valve and the speed-control mechanism, the attendant should feel very much in command of the test.

Means for priming the pumps with ease and certainty should be thought out, § 292 ; and the unexpected talent of works testing tanks for collecting rubbish --wood, cotton-waste, etc. —must not be overlooked. Water continuously circulated soon grows foul, and moreover the grease it carries with it might sensibly falsify the indications of gauging weirs unless these are regularly cleaned with petrol.

One remaining matter is the layout of the log-sheets on which observations are entered during the test, and of the final forms recording the pump's performance suitable for filing or for handing to the client. Linked herewith is the provision of all needful help—tables, charts, etc.—for the computing staff.

(Example 22)

Tests on Site

178. Some Comparisons. There may be at least two reasons for carrying out running tests on a pump after it has been erected on site. (i) Acceptance tests may be essential because the pump was too big or was otherwise unsuitable for the shop testing-plant. (ii) In the course of the normal operation of the pump, periodical checks on its performance will show when overhauls and possibly repairs or complete replacements are indicated, and these tests may thus avert excessive power costs resulting from impaired efficiency. Under the best conditions, testing the complete pumping installation when finally assembled will appear relatively easy. The installation will include pressure-gauges for suction and delivery branches, control valves in plenty, and permanent Venturi meters or the like for measuring flow, § 295. On the other hand, the facilities for calibrating the instruments on the spot may be absent, it will almost certainly be impossible to measure directly the power input to the pump, and it is improbable that the engineers in charge will be able to establish pre-determined heads and rates of flow. For example, a borehole pump will have to work against the head as fixed by the prevailing underground water conditions ; a low-lift irrigation pump will have its suction lift settled by the momentary level of the river from which it draws. Service conditions in the system as a whole may prohibit a long unbroken test run at a fixed rate of discharge.

Under the worst conditions there may be no permanent measuring gear of any kind. Instruments of every sort must either be brought to the site or improvised there. Intermediate in character are those tests of very large pumps that are provided with water-level or pressure gauges, but have no flow-measuring equipment.

Typical of the most favoured acceptance tests are those organised at an important new waterworks pumping plant. No thought or time or labour has been spared to ensure an installation of first-class quality in every way. The pumps have already been run in, and nursed through such initial troubles as are likely to occur. An array of technical talent will watch over the actual trials, where one may expect to see contractors' representatives in company with the engineer who has designed

the plant and who will be responsible for its operation. The assembly should be in little need of outside advice.

179. Flow-measurement on Site. When there is no permanent metering system it is again possible to differentiate between two possibilities. The mere absence of equipment may suggest that the plant is itself inexpensive or relatively unimportant, from which it follows that only an inconsiderable sum of money will be available for making tests. But expense need be studied less rigorously when making trials of the pumps destined for graving-docks, or for irrigation, drainage, or hydraulic-storage installations. Here the discharge per unit may amount to 10 tons of water per second, and the power input may be anything from 500 h.p. to 10,000 or even 50,000 h.p.

If the pump output is small there may at least be a chance that a temporary measuring basin can be contrived; thus, a short length of canal might be dammed off, or some old boiler or disused tank pressed into service. For continuous flow-measurement, two new methods have lately been developed, the pipe-bend meter (*) and the suction-pipe meter (*), whose cost is almost negligible.

Among large-capacity installations, dock-pumps and hydraulic-storage pumps are very fortunately situated. The graving-dock itself, or the accumulating basins of the storage system, should serve as excellent volumetric measuring tanks, although it is true that the tests must be run under those conditions of variable head that were mentioned in § 172. For other low-lift pumps there is a possibility of rigging up a temporary weir, but it may be difficult to ensure those standardised flow conditions which alone make the weir gaugings acceptable. The instruments that remain are the Pitot-tube or its fellow, the Pitot sphere; and the current meter. Pitot-type instruments must almost necessarily be used in closed pipes, for only there is the velocity likely to be high enough to generate a differential head that can accurately be measured. Current-meters, at least of the propeller type, have been successfully used in the pump suction pipe, or the pump delivery pipe, or in the open channels that lead the water to or from the pump. The system of multiple meters developed for hydraulic turbine tests has also given good results in pump tests.

Pump Faults and their Correction

180. Some Possible Faults. Defects that may be revealed during any type of test, or after the test results have been analysed, may include :—

(i) Pump runs hot or runs noisily.
(ii) Pump generates incorrect head when delivering stipulated discharge at stipulated speed.
(iii) Pump absorbs excessive power.

Faulty Running. Overheated *glands* may be due to uneven or excessive tightening of the gland nuts. Overheated *bearings* may result from such obvious causes as mis-alignment, distortion of frame, etc., or else from some internal hydraulic irregularity, e.g., partial choking of one eye of a double-inlet impeller, § 76. *Vibration and noise* may be caused by imperfect mechanical balance of the rotating parts, or by cavitation resulting from excessive suction lift, or by unsuitable blade design, or by lack of stiffness in general construction.

Incorrect Head. Excessive suction lift, poor axial location of the impeller on its shaft, or partially-choked rotor or diffuser passages may likewise prevent the pump from generating its designed head. It is not inconceivable that the impeller might have been mounted wrong way round on the shaft.

Excessive Power Consumption. If this is not accompanied by excessive head, it may be due to abnormal internal friction. For example, the sealing-ring clearance, § 83, may have been cut too fine, resulting in actual metallic contact between fixed and moving parts. In a multi-stage pump with balancing disc, § 124, partial choking of the balance-water drain may make the system inoperative, and again there will be metallic rubbing contact. In any event heat and noise and vibration and excessive energy loss usually go together.

181. Possibilities of Correction. By checking and correcting the instruments of the test-plant ; by draining, stripping, and correctly re-assembling the pump ; it may be possible to record acceptable test figures when the pump is again put on the test-bed. But if the behaviour is still not up to standard, there is only one main type of defect that can be remedied without making new components. If the head generated is too *high*, then the impeller blade tips can be turned down slightly in the lathe, Fig. 117. The corrected diameter can be roughly computed thus:—

If H_e = specified head,
H_{et} = observed head on test,
d_{2t} = diameter of impeller on first test,
d_2 = desired corrected diameter.

Then $d_2 = d_{2t}\sqrt{\dfrac{H_e}{H_{et}}}$.

The impeller *shrouds* or discs should not be touched: the original diameter should be left undisturbed. This procedure is so simple and effective that it is a common workshop custom.

A still simpler modification is this: the blade tips can be filed to a taper form, thus slightly reducing the effective outlet angle, and therefore lowering the head.

Screw-type rotors are not amenable to either of these types of corrections; but axial-flow pumps of suitable types may be regulated by adjusting the blade angles, Fig. 103.

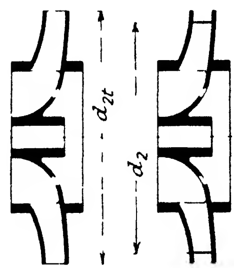


FIG. 117 —Reduction of impeller diameter.

SPECIAL TESTS

182. Purpose of Special Tests. One might describe as "special" any type of experimental investigation other than routine testing of a pump destined for sale or stock or actually installed on site. Such special tests may include :—

(i) An actual pump intended for service may be tested with *air* as the working fluid, instead of with water or other liquid.

(ii) Specially-built scale model pumps may be tested either with liquid or with air, in order to predict the performance of the full-size prototype.

(iii) On full-size or on model pumps, measurements and observations of various kinds may be made with the object of studying internal flow conditions; these are additional to the basic measurements of head, discharge, power, and speed.

183. Testing with Air. As mentioned in §§ 171, 177, there are two reasons why the testing of large pumps may be expensive: both the cost of the special test rig and the cost of

the power absorbed during the test may be quite appreciable. By using air instead of water, § 236, the whole procedure may be much simplified and expenditure correspondingly reduced. But these tests with air are not a complete substitute for the normal routine of testing with water. They give no reliable information about the power consumption of the pump during working conditions. So the technique– which is still in course of development– seems to be attractive chiefly when preliminary trials are to run off. After tests with various types of rotor have shown which shape gives the desired results, then only the final confirmatory or acceptance test of the pump need be run with water as the working fluid.

The advantage of testing *scale models* with air is that it gives the opportunity of controlling the density of the working fluid and thereby collecting information which assists in interpreting the results of scale model tests in general, § 237.

The *test equipment* required for any kind of air tests at atmospheric pressure is particularly simple. For measuring pressure, sensitive water gauges of the type used for draught indication will serve ; or a differential gauge as in Fig. 116 (iv) can be contrived. For measuring discharge, standard flow-nozzles set in a straight length of delivery pipe should give good results.

When testing models with *compressed air*, some kind of closed-circuit arrangement on the lines suggested in Figs. 114, 115 will be necessary.

184. Scale Model Tests. Here we return to the more common problem of testing scale-model pumps with water or other liquid. Although in general routine information concerning head, speed, discharge, etc., will serve ordinary needs, we have to remember that the observations must now be made with unusual precision. A good deal of preliminary study is required in fixing the *scale* of the model. The bigger the model, the more reliable will be the observations made upon it ; but the greater also will be the cost of construction and the cost of the power consumed during the tests– nor can it be forgotten that the tests may be quite protracted.

It is particularly in measuring the *power input* to the model that the test-bed routine must be scrutinised. What the tests are designed to show may be not so much the absolute efficiency

of the model as the small changes in efficiency produced by small changes in construction, e.g., variations in rotor shape, blade angle, inlet passages, etc. Now in small pumps such as the model we are talking about, the friction of bearings and stuffing-boxes is highly susceptible to what seem to be trifling influences. It may be found, for instance, that during the course of an hour's run under nominally unchanged conditions, the pump efficiency has apparently risen by 1 per cent., solely due to the natural warming-up of the ring-lubricated bearings and the resulting drop in the viscosity of the lubricant. At least equally serious anomalies might result from inattentive tightening of the glands.

Results of tests of model pumps are computed and utilised as explained in §§ 224 to 226.

185. Observations of Internal Flow Conditions. The Bibliography at the end of the book, p. 481, must serve to show what a great range of research on this matter has already been carried out. Without going into details, one can nevertheless discern a few broad lines which investigators have followed, e.g.: -

(i) Instruments such as Pitot tubes or Pitot spheres, inserted through holes drilled at various points in the pump casing, have revealed the magnitude and direction of the absolute velocity components at selected stations within the pump, § 207 (ii).

(ii) Lightly pivoted vanes with external indicators or pointers have shown velocity direction, such as the direction of the liquid leaving the impeller.

(iii) Pressure gauges of various kinds have recorded pressure variations across selected planes.

(iv) High frequency oscillographs, in conjunction with sensitive pressure-responsive elements, have recorded more rapid pressure fluctuations.

(v) Experimental pumps made of transparent material or provided with observation windows have permitted visual inspection of phenomena within the pump. To make the liquid streams visible, injected coloured dyes or small floating balls have served. Intermittent illumination on the stroboscopic principle has enabled observers to study the bubbles formed during cavitation.

CHAPTER XIII

PERFORMANCE UNDER DESIGN CONDITIONS

186. The Basic Enquiry. Various paragraphs on pump design in Part B of the book have presented empirical information which enabled the efficiency of a selected type of pump to be predicted. The test-bed figures will show whether or not those estimates were warranted. There is no way of escaping wholly from this trial-and-error method ; but the more closely the details of pump behaviour are analysed, the smaller should be the discrepancy between promise and performance. It is the purpose of this chapter and the following ones to attempt this analysis.

The test-bed results have proved that energy at a rate S.H.P. must be fed into the pump shaft ; yet the liquid only receives energy at a rate W.H.P. What has happened to the energy that has disappeared ? That is the basic question. In a general way we can discern four main possibilities of energy dissipation : –

(i) Mechanical friction between fixed and rotating parts in bearings and stuffing-boxes.

(ii) Hydraulic friction— so-called disc friction—between the liquid and the external rotating faces of the rotor discs.

(iii) Wherever there is leakage of liquid, energy will be dissipated.

(iv) The main flow of the liquid through the passages of casing and rotor will be subject to hydraulic loss due to eddying and friction.

The input power, the output power, and the power dissipated in these various directions may be denoted thus :—

	Input or shaft horse-power = S.H.P.	$= P_s$.
	Output or water horse-power = W.H.P.	$= P_w$.
(i)	Mechanical friction power loss	$= P_b$.
(ii)	Disc friction power loss	$= P_d$.
(iii)	Leakage, slip, or short-circuit power loss	$= P_l$.
(iv)	Hydraulic power loss	$= P_h$.

$$\text{Gross power loss} = P_b + P_d + P_l + P_h = P_t.$$

They are suggested symbolically in Fig. 118.

187. Classification of Power Losses. One system of grouping these losses is into the categories of mechanical and hydraulic; class (i) might be termed mechanical, but all the others are in a sense hydraulic. But another system is found preferable from the point of view of energy accountancy; it permits a clearer and more easily understood energy balance sheet to be drawn up. The separation is on these lines :—

(*a*) Losses which affect the main liquid stream flowing through the pump, viz. P_h.

(*b*) All other losses, viz. P_b, P_d, P_l, whose sum can be denoted P_{bdl}.

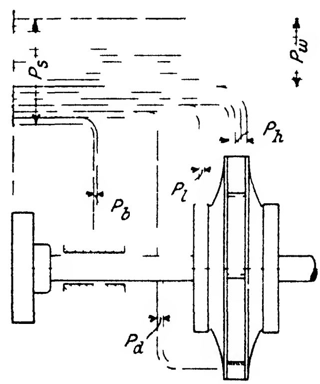


FIG. 118.—Diagram showing where power is dissipated

For a given rotational speed, category (*a*) will be directly dependent upon the rate of flow of liquid or the discharge of the pump, while category (*b*) will be sensibly unaffected by the rate of flow. This distinction will be of great utility in the conditions of varying flow studied in Chapter XIV. Moreover, the distinction is logically linked with the definitions of efficiency framed in Chapter XI. Only losses of type (*a*) can influence the *hydraulic* efficiency of the pump, § 164; but all the losses might influence the *gross* efficiency, § 165. Although we must recognise that the distinction is arbitrary, it is nevertheless effective.

As for the method of presenting information about each individual loss, the non-dimensional system proposed in Chapter V will be equally useful here. We may be interested not so

much in the absolute value (say) P_d of the power loss, as in the *relative* value: the ratio between the power loss and the gross power input P_s, viz. P_d/P_s. Convenient symbols are:—

Relative mechanical loss $= P_b/P_s = \Delta_b$.
Relative disc friction loss $= P_d/P_s = \Delta_d$.
Relative leakage loss $= P_l/P_s = \Delta_l$.
Relative hydraulic loss $= P_h/P_s = \Delta_h$.
Relative gross loss $= P_t/P_s = \Delta_t$.
Relative loss $P_{bdl}/P_s = \Delta_b + \Delta_d + \Delta_l = \Delta_{bdl}$

Since $P_u = P_s - P_b - P_d - P_l - P_h = P_s - P_t$, and $\eta_m = P_u/P_s$, § 165, evidently

$$\eta_m = \text{gross efficiency} = 1 - \Delta_t \quad . \quad . \quad (13\text{-}1)$$

In this chapter the general nature of the various losses will be examined, and the manner in which the *shape number* is likely to influence them, *when the pump is running at designed speed, head, and discharge*. Later chapters will study the effect of change of discharge, change of speed, change of liquid, change of scale, etc.

188. Mechanical Friction Loss. The components of the pump that may here be involved are:—

The external sleeve, ball, or roller journal bearings.
The internal sleeve bearings.
The thrust bearing.
The gland, neck-bush, and packing-rings.

Now the proportion of the total mechanical loss contributed by each of these will manifestly depend very much upon the type and condition of the pump. In a propeller pump we might expect the thrust bearing to absorb a relatively large share of the energy, while in a double-suction split casing centrifugal pump in good order the thrust collar should have virtually nothing to do. In a belt-driven centrifugal pump such as in Fig. 39 (iii) it might be the journal bearings that are wasteful of energy. The most variable item of all is the stuffing-box friction: sometimes there may be one stuffing-box, sometimes two, and in each box the torque to be overcome will be influenced by the type and condition of the packing, the pressure acting upon it, and the degree of tightening of the gland nuts.

But whatever may be the combined effect of these factors,

the resulting value of the relative friction loss $\wedge_b$ is never very high: it may vary between the limits

$$\wedge_b = 0{\cdot}01 \text{ to } 0{\cdot}04.$$

It is thus almost invariably smaller than any of the other relative losses.

189. Disc Friction Loss. In principle the parts concerned here are all those rotating surfaces of the pump in contact with the liquid that do not actually take a share in guiding the liquid. But as such elements as the shaft make so very small a contribution to the total loss, we may disregard them and concentrate on:—

The external faces of the rotor discs or shrouds.
The outer edges of the shrouds.
The edges of the sealing-rings.
The whole surface of the balance-disc, if fitted.

These are indicated by broken lines in Fig. 119 (i).

Evidence on which to base estimates of the power loss P_d comes primarily from experiments on flat discs each rotating in a concentric casing filled with liquid, as in Fig. 119 (ii). By measuring the power needed to drive round the disc in a variety of conditions, it appears that this power loss depends upon:—

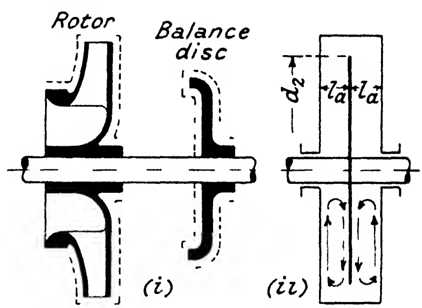


FIG. 119 Diagram showing areas subject to disc friction loss, etc.

(*a*) The rotational speed in r.p.m., viz. N.
(*b*) The disc diameter, d_2.
(*c*) The roughness of the sides of the disc and the inner walls of the casing.
(*d*) The density w and the viscosity μ of the liquid.
(*e*) The axial clearance l_a between the disc and the casing.

The variables (*a*) to (*d*) are comparable to or identical with those that influence flow through closed conduits. As for (*e*), the axial clearance affects the power loss for the following reason: we may assume that any element of liquid in contact with the revolving disc will be dragged round with it, at least

for a short distance; and during this journey the element will necessarily be subjected to centrifugal force. This will induce it to slide outwards. Other elements from the main body of liquid will flow in to replace the original one, and hence a kind of circulation will be set up as indicated in Fig. 119 (ii). In other words, frictional impulsion has created on a very small scale the pumping effect that the direct thrust of the impeller blades creates on an effective scale.

We have to regard this secondary circulation as being superimposed upon the tangential motion described in § 74. But it seems likely that in the space between disc and casing, the axial distance l_a in Fig. 119 (ii) will influence both the radial and the tangential velocity components. If this distance is large, it will be easy for relatively large amounts of liquid to become involved in the secondary circulation and thereby to steal energy from the disc; but if the distance is small, the energy loss should be less. Experiments confirm this: except for minor irregularities, the rule is that in comparable conditions an increase in the axial clearance l_a causes an increase in the power loss P_d.

190. Estimating the Disc Friction Loss. From an analysis of experimental results based on water as the liquid, a first approximation to the value of the disc friction loss P_d can be put in the form

$$P_d = K_{df}\left(\frac{N}{1000}\right)^3 d_2^5 \quad . \quad . \qquad (13\text{-}2)$$

where K_{df} is a coefficient having a mean value of about 0·37 when d_2 is expressed in *feet*, and P_d expresses the power dissipated on *both* sides of the disc (*).

A more general expression, having the structure analogous to that which serves for flow formulæ in smooth pipes, would have the form

$$P_d = K_{dfs}\,(R_{nd})^{0.15} w v_2^3 d_2^2 \quad . \quad . \qquad (13\text{-}3)$$

where K_{dfs} is a coefficient having an invariable value for *smooth* discs, v_2 is the rim velocity, R_{nd} is a type of Reynolds number represented by $\left(\frac{v_2 d_2}{\nu}\right)$, and ν is the kinematic viscosity of the liquid.

Even if an authoritative value for K_{dfs} could be established, there still remains the task of accounting for : —

(1) The roughness of the disc and the casing walls.

(2) The effect of the axial clearance l_a, § 189.

(3) The effect of the diameter ratio d_1/d_2, § 92. In double-inlet impellers the shrouds do not form complete discs but only annular surfaces ; and as the diameter ratio d_1/d_2 increases, so may the power loss for a given value of d_2 be expected to fall.

(4) The effect of a tertiary flow on the motion of the liquid between discs and casing. Liquid leaking from the recuperator back to the impeller eye will still further complicate the motion described in § 189. At the moment of leaving the impeller an element of such liquid will be moving with the full tangential velocity V_n, and therefore at first it will have small chance of abstracting energy from the impeller discs.

(5) The viscosity of the liquid if laminar conditions prevail. Even if the liquid were not so viscous as to modify radically the general form of equation (13-3), its viscosity might very seriously affect the regime in the narrow clearance *between the fixed and rotating wearing rings*, § 84. Thus if the labyrinth type of seal, Fig. 48 (x), had been chosen, and a heavy oil were passing through the pump, we can imagine what an effective kind of brake the sealing-rings would form.

(6) The power loss associated with the unsupported edges of open-type impellers and of all types of screw and axial-flow rotors. This may more conveniently be included under the heading of leakage loss, § 191.

At this stage, then, we cannot do more than say that the value of the relative disc friction loss may vary within the range

$$\triangle_d = 0{\cdot}02 \text{ to } 0{\cdot}08.$$

191. Defining the Leakage Power Loss. According to § 82, the volume of leakage liquid q_l that slips past the sealing-rings of a centrifugal pump may be of the order of $(0{\cdot}03 \text{ to } 0{\cdot}10)Q$, where Q is the net flow through the pump. This information could be expressed in another way. We might introduce the notion of *volumetric efficiency*, defining it as the ratio $\left(\dfrac{Q}{Q+q}\right)$. But that ratio as it stands would not necessarily tell us how

the leakage power loss is related to the pump water horsepower. The reason lies here: that while the main flow Q passes through the rotor *and* through the recuperator, the leakage flow q_l is diverted *before* it can benefit from the recuperator. Because of this, and because of further difficulties that might arise, § 192, it is preferable to drop this concept of volumetric efficiency and to make a more radical distinction between main flow Q and leakage flow q_l. We are to suppose that the rotor has an imaginary annular partition which divides it into two entirely distinct passages, a main passage for the main flow Q and an auxiliary passage for the leakage flow q_l, Fig. 120 (i). If imaginative readers prefer to visualise a little independent rotor for handling the leakage flow, Fig. 120 (ii), so much the better. Henceforth we can think of the main flow as entering the pump at the suction flange, passing unchanged through rotor and recuperator, and finally issuing at the delivery flange. Meantime the leakage flow is continuously circulating through its own auxiliary rotor and back through the sealing-rings.

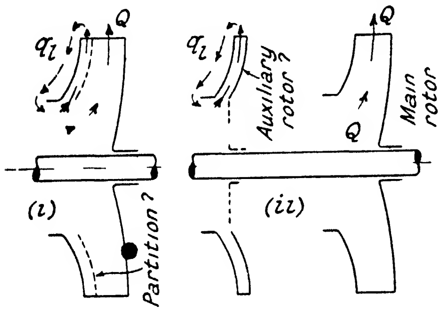


FIG. 120 Diagram showing how leakage flow can be distinguished from main flow.

Now the entire energy required to maintain the leakage flow must be drawn from the pump shaft, and it is the corresponding power loss P_l that we now wish to estimate. First the number of leakage passages and the leakage areas must be defined as in § 83, then the pressure-drop H_l, and in turn the leakage volume q_l. The W.H.P. of the auxiliary or "leakage" impeller will then be $q_l w H_l / k_p$. Now the gross efficiency of the auxiliary impeller must necessarily be *lower* than that of the main impeller, because there is no auxiliary recuperator, as has just been pointed out. By applying a suitable correction, we finally arrive at the S.H.P. of the leakage impeller, viz. the leakage power loss P_l.

192. Assessing the Leakage Power Loss. The pro-

cedure just outlined is sufficiently beset by uncertainties even for the simplest case of a radial-flow impeller. But when examining a general problem we must realise that the leakage liquid does not all flow through a single passage but is " bled " off the main stream at a number of points, each possibly at a different pressure. Leakage may occur at other regions than at the sealing-rings, e.g. :—

(*a*) Through the lantern rings of liquid-sealed stuffing-boxes, § 85 (ii).
(*b*) Past the balance-disc of multi-stage pumps, § 124.
(*c*) Past the neck-bushes in the diaphragms of multi-stage pumps, § 122.
(*d*) Past the main glands to waste.

There remains the highly complex question of leakage past the unsupported edges of screw type and propeller-type rotors, § 100. It is true that the narrow clearance space between the blade edge and the fixed casing constitutes a sort of passage ; it can be admitted that the rate of flow through the passage will be influenced by the pressure difference across it. Yet, remembering that this pressure difference probably varies from point to point along the blade edge, and observing that one side of the passage is at rest while the other side is travelling round with peripheral velocity v, we can admit also the futility of applying the simple laws of orifice flow, § 83. Moreover, we have already agreed, § 190, to include in this component of the leakage power loss the hydraulic friction loss between blade edge and casing.

In a normal pump without special measuring gear, only item (*b*) of the leakage power loss, that associated with the balance-disc, can be estimated at all accurately. It may be possible to measure the volume of leakage water as it flows to waste ; and as the pressure-head at the leak-off point is fairly well known, this particular item of the total loss P_l can be computed by the method of § 191. **(Example 20)**

As for the overall value, a rough indication must again suffice. The relative leakage power loss Δ_l may vary from about

$$\Delta_l \quad 0{\cdot}03 \text{ to } 0{\cdot}12.$$

193. The Hydraulic Power Loss. If we have correctly

assessed the respective values of the mechanical loss P_b, the disc friction loss P_d, and the leakage loss P_l, then we can assert that the residual energy $P_s - P_b - P_d - P_l$ must wholly be transferred to the main stream of liquid flowing through the pump. There is nowhere else for it to go. But this is a very different thing from saying that the liquid receives a corresponding net energy increment during its passage from suction flange to delivery flange. The difference between the two quantities is what we have agreed to call the hydraulic power loss, P_h, and it can be computed—at least in principle—in this way :—

The energy E_n transferred to the liquid is utilised in imparting tangential acceleration to the liquid elements. Unit weight of liquid will receive $\frac{V_n v_2}{g}$ units of energy, and as there are W units of liquid effectively flowing per second, we can say that

$$\frac{W}{k_p} \cdot \frac{V_n v_2}{g} = P_s - P_b - P_d - P_l = P_w + P_h \quad (\S\S\ 11,\ 24).$$

Now the *net* energy increment per unit weight of liquid is represented by the effective head H_e, § 164, and the *net* power received, W.H.P. or P_u, is $\frac{WH_e}{k_p}$.

Therefore *hydraulic power loss* P_h can be written

$$P_h = \frac{W}{k_p}\left(\frac{V_n v_2}{g} - H_e\right),$$

and the *hydraulic efficiency* η_h can be written, § 164,

$$\eta_h = \left(\frac{H_e}{V_n v_2/g}\right) = \frac{P_w}{P_u + P_h}.$$

Unfortunately these expressions give very little help in yielding an actual numerical value for the power loss P_h, because *we do not precisely know the value* of the true whirl component V_n. It is just these equations which serve to establish the value of V_n.

194. Analysing the Hydraulic Loss. Inconvenient though it may be, the fact must be accepted that although the hydraulic power loss is nearly always the most serious of any in the pump, yet it is the one that can be assessed with the least

accuracy. From the *measured* input power P_s we must subtract in turn the *measured* output power P_w and the *estimated* losses P_b and P_d and P_l. Clearly, therefore, the final difference P_h may be burdened with the cumulative errors affecting the other terms of the equation; and earlier paragraphs have indicated how wide a margin there may be here for such individual uncertainties. But at least the representation of the hydraulic loss can be done in a familiar way, especially if we express the loss in terms of *energy per unit weight*, e.g., foot pounds per pound. The complete pump can be regarded as a rather complex closed conduit, and the energy changes as the

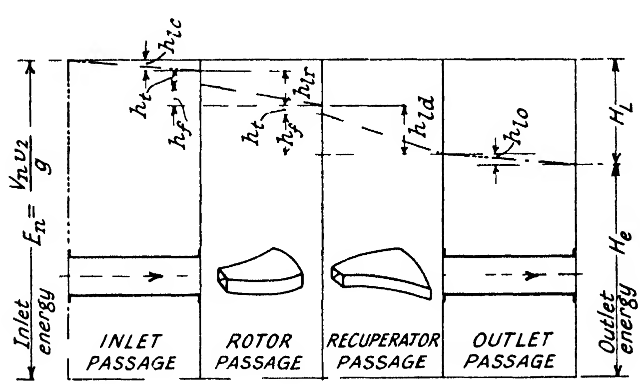


FIG. 121 Hydraulic losses in pump passages.

liquid flows through this series of passages can be plotted as a simple graph. If in the first place we disregard the energy received from the rotor, then what we are now seeking is merely a corrected form of the basic diagram Fig. 4. Then when we want to show the energy increment it involves no more—in principle—than elaborating the energy diagram in Fig. 108. That diagram merely stated in a purely conventional way that the liquid received an amount of energy H_e while passing from the suction flange to the delivery flange of the pump; now we must separate that net gain into a gross intake of energy and into various kinds of energy dissipation; we must plot to an enlarged scale the part of the diagram included between the pump flanges.

The complete conduit or series of passages now to be studied is shown schematically in Fig. 121. It comprises (i) the inlet

part of the casing, from suction flange to rotor inlet, (ii) the rotor passages, (iii) the recuperator passages, (iv) the outlet part of the casing, from recuperator outlet to delivery flange. Although it may usually happen that no physical separation can be discerned between items (iii) and (iv), it is proper that in a quite general statement we should recognise the possibility of such a division. In all the passages except the rotor passages, (ii), it is inevitable that energy losses will occur (*).

But how shall we deal with the special conditions in these revolving rotor passages? At the same time that the liquid is very rapidly *receiving* energy from the rotor blades, it is *losing* energy by frictional contact with the walls of the passages. The solution proffered in Fig. 121 is this: let us assume that the gross energy increment $E_n = \frac{V_n v_2}{g}$ is imparted to the liquid not while it flows through the rotor passages, but in the form of a sudden "shot" or dose just as it enters the pump. This will in no way interfere with the final result, and it puts the rotor passages on the same basis as all the other passages. The diagram can now be regarded as an alternative and corrected form of Fig. 8.

195. Interpreting the Energy Diagram. The proportions of the energy diagram, Fig. 121, at once give a measure of the *hydraulic efficiency* of the pump, § 164. The energy input per unit weight of liquid is represented by the distance

$E_n = \frac{V_n v_2}{g}$; the energy output by the distance H_e or effective head; thus the hydraulic efficiency is represented by the ratio $\eta_h = H_e/E_n$. Also if the total energy loss, $E_n - H_e$, is denoted by H_L, then the hydraulic *power* loss P_h will be expressed by

$$\frac{WH_L}{k_p}, \text{ § } 193.$$

Here are a few comments which serve to link up earlier statements with the present discussion, as symbolised in Fig. 121.

(i) *Inlet Passage.* Evidently the design of the casing will exert the controlling influence on the energy loss, h_{ic}. In side inlet pumps, Figs. 38 and 40 (ii), the loss may be virtually zero: in double-inlet split-casing pumps, Fig. 39, the loss will be quite

appreciable, but it can be reduced if necessary by the special form of casing shown in Fig. 51 (i).

(ii) *Rotor Passages.* The greater the number of blades, the greater will be the area of metallic surface which can impose frictional loss on the liquid. The smaller the outlet angle γ, the longer and narrower will be the rotor passages, which again means a greater frictional loss, h_f. The total loss h_{lr}, Fig. 121, may include an item h_t to cover eddy, shock, or transfer losses at the blade inlet edges. At part flow this item may be of serious consequence, §§ 204, 205.

(iii) *Recuperator Passages.* The energy losses here, h_{ld}, are all of the general type examined in §§ 43 to 45. They are represented in the diagram by an entry or eddy loss h_t and a friction loss h_f.

(iv) *Outlet Passages.* Multi-stage pumps provide examples of outlet passages that are functionally distinct from the recuperator passages. In the return passages of the diffuser disc, § 122, the energy loss. h_{lo} is additional to the unavoidable energy-transformation loss in the diffuser passages themselves. That is why in this class of pump the combined energy loss due to items (iii) and (iv) is unusually high.

All these energy losses will vary roughly as (velocity)^2. To sum up, then, we can say that the overall hydraulic *power* loss P_h depends generally upon the design of the passages and probably upon the *cube* of the velocity of the liquid flowing through them. The corresponding relative power loss $\triangle_h$ may range from about

$$\triangle_h = 0{\cdot}06 \text{ to } 0{\cdot}15.$$

Relation between Shape Number and Energy Losses

196. Influence of Change of Shape Number. From the foregoing preliminary examination of the general nature of the relative power losses in rotodynamic pumps, we cannot extract very much useful quantitative information—the figures quoted vary through far too wide a range. Nor for the moment is there much hope that we can learn to define the respective values of $\triangle_b$, $\triangle_d$, $\triangle_l$, $\triangle_h$ with sufficient precision to enable us to make a confident forecast of the pump efficiency. But what we can

do, and what we forthwith proceed to do, is to find out how *changes* in pump design are reflected in *changes* in pump performance. If absolute values are as yet unattainable, then comparative values may be quite useful.

As in other instances, the most instructive manner of displaying variations in pump design (*) is to show how the value of the shape number or of the specific speed affects the other variables that interest us, §§ 60, 61. Dealing first with (nominally) radial-flow centrifugal pumps, we observe from § 59 that their shape numbers can be put in the form $950\ \phi\sqrt{\lambda\psi}$. In practice, successive changes in rotor shape are brought about by simultaneously altering the speed ratio ϕ, the flow ratio ψ, and the width ratio λ, as in Fig. 54. But if we prefer, there is nothing to stop us from changing these ratios one at a time, in order to observe more positively the corresponding effect on pump efficiency. Although it may turn out that some of the resulting rotor shapes would have little practical utility, yet in every case we shall assume that they are properly suited for working under " design " conditions.

197. Influence of Width Ratio (*). We begin with a range of pumps in which everything remains unchanged except the rotor width and the variables dependent upon rotor width, thus :—

Invariable.	*Variable.*
Speed.	Width.
Diameter.	Width ratio.
Blade angles.	Discharge.
Speed ratio.	Volute area.
Flow ratio.	
Head.	

By making a preliminary plot of power and power losses against width or discharge, we get an array of graphs of the sort shown in Fig. 122. The respective scales are purposely distorted for the sake of clarity. In this diagram and in succeeding ones, the symbols are those defined in §§ 186, 187. Evidently the mere act of moving the impeller discs further apart can have no effect upon the losses P_b, P_d and P_l, and thus the graph representing the sum of these is a horizontal straight line. Furthermore, we may reasonably assume that changes in

the rotor width will not materially alter the proportions of the energy diagram, Fig. 121. It follows that the *hydraulic efficiency* η_h remains unaltered throughout, and that the absolute hydraulic power loss P_h varies directly as the discharge or as the output power P_w. In turn the *relative* losses and the efficiencies can be plotted against shape number as in Fig. 123; this true-to-scale diagram is based upon the values $\phi = 1{\cdot}00$, $\psi = 0{\cdot}10$. Types of the rotors themselves are drawn to scale in Fig. 124. **(Example 21)**

We may take Fig. 122 as a graphical comment on the principle set forth in § 187; we clearly see how advantageous it is

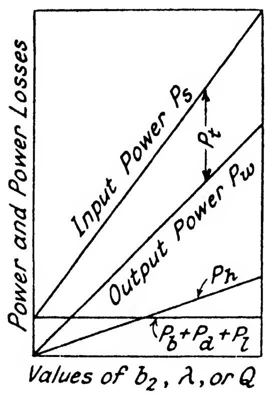


FIG. 122 Influence of impeller width on performance

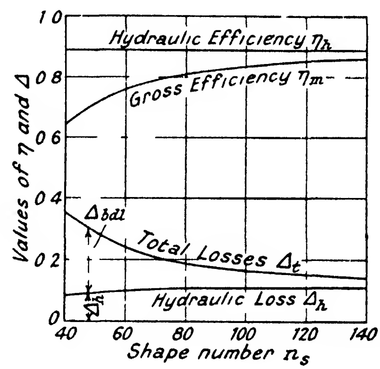


FIG. 123 — Relationship between width ratio and efficiencies

to separate losses dependent upon the flow, P_h, from those unaffected by the flow, P_b, P_d, P_l. As for the trend of the gross efficiency curve in Fig. 123, it indicates that the efficiency would be zero at zero specific speed, and that it approximates to the value of the hydraulic efficiency at high specific speeds. But we have to remember that at high specific speeds the flow conditions in the rotor become in fact less favourable, because of the eddies shown in Fig. 124. This deterioration will be reflected in both efficiency curves of Fig. 123, which will tend to droop in the region of high shape numbers.

198. Influence of Flow Ratio. Here the fixed and varying attributes of the range of impellers are :—

Invariable.	*Variable.*
Speed.	Blade angles.
Diameter.	Velocity of flow.
Speed ratio.	Flow ratio.
Width.	Discharge.
Head.	
Flow areas.	

As before, the "non-hydraulic" losses P_b, P_d, P_l remain unchanged; while the hydraulic power loss P_h changes. But now there is a different law of variation. Since the energy loss varies as the *square* of the flow velocity, then the power loss varies as the *cube* of the discharge, § 195, instead of *directly* as the discharge as in § 197. The effect on the graphs, Fig. 125,

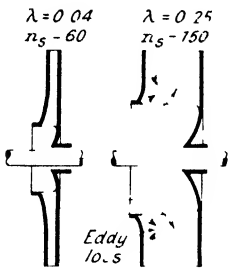


FIG. 124. Connection between impeller width and shape number.

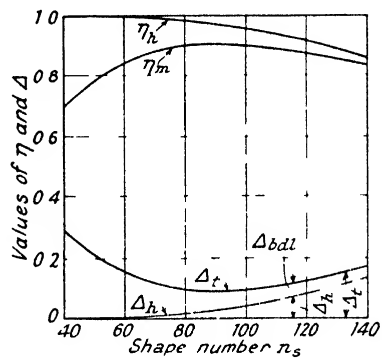


FIG. 125.—Effect of flow velocity on efficiency.

is very marked—there is no doubt now about the decline in efficiency at high values of shape number. In this diagram the data are: constant speed ratio $\phi = 1{\cdot}0$; constant width ratio $\lambda = 0{\cdot}10$.

199. Influence of Speed Ratio. The factors controlling this range of rotors are:—

Invariable.	*Variable.*
Diameter.	Blade angles.
Width.	Speed.
Discharge.	Speed ratio.
Head.	
Flow areas.	

The trends to be examined now are those that were already exposed in a primitive way in § 14. It is still true, as Fig. 9 showed, that as the outlet angle γ *increases* and the speed ratio *falls*, the liquid leaving the impeller carries away with it a greater and greater proportion of velocity energy. We now see that this tendency throws an increasing burden on the recuperator. In turn this means that we must make some specific estimate of the energy loss in the recuperator, h_{ld}, Fig. 121. Let us say that this amounts to one-fifth of the *ideal* velocity energy at the impeller outlet, viz. $h_{ld} = 0{\cdot}20 \dfrac{U_2^2}{2g}$. In regard to the energy loss in the impeller passages, this may be taken to be proportional to the *square* of the outlet relative velocity v_{r2}.

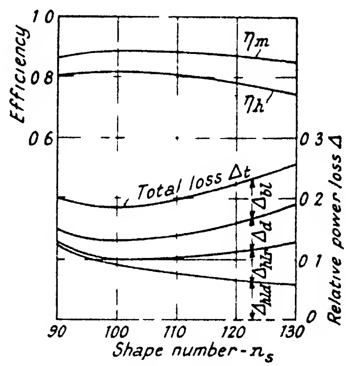


Fig. 126.—Influence of speed ratio on performance.

For the first time, then, we have under review a set of impellers in which the ratio between impeller energy loss and recuperator energy loss is varying. The character of the variation is suggested in Fig. 10; since impeller loss varies as $(v_{r2})^2$, and recuperator loss varies as U_2^2, we see at once that increasing values of speed ratio mean an increasing proportion of *impeller* loss. Another new factor is that the disc friction loss is no longer constant: it varies as the cube of the speed, § 190. The remaining losses P_b and P_l will only change very slightly. When these various influences are taken into account, and applied to impellers having the ratios $\psi = 0{\cdot}15$ and $\lambda = 0{\cdot}10$, the resulting graphs have the forms shown in Fig. 126. Although the range of shape number is less than it has hitherto been, yet the efficiency graph keeps its characteristic form showing a marked decline at either extreme.

Naturally we cannot accept the graphs as a final statement. On looking again at the ideal velocity diagrams in Fig. 10, we cannot profess any great faith in the belief that impeller losses vary as the square of the mean velocity in those passages.

In *diverging* passages we know that it is the angle of divergence as well as the mean velocity that determines the energy loss, and at least in Fig. 10 (iii) the divergence is so quick that we should expect very unfavourable hydraulic conditions there. On the other hand if we go to the other extreme, and make the passages even longer and narrower than they are in Fig. 10 (i), we cannot hope for any mitigation of the serious friction losses that will ensue, § 195 (ii).

200. General Trends. What, then, is the precise value of graphs such as those presented in Figs. 123, 125, 126 ? They are admittedly based on simplified and highly artificial conditions, which only in a very general way accord with actual working conditions. But at least the trends they reveal are fairly clear. We can discern two of them : (i) the pump gross efficiency rise to a maximum figure at a certain value of the shape number, and declines on either side of that number, (ii) as the shape number increases, the *hydraulic* power loss grows more and more preponderant. That is to say, the ratio Δ_h/Δ_t has a rising tendency. If, then, we were to construct a set of composite graphs, applicable to a range of normal rotors in which the terms ϕ, ψ, and λ increase simultaneously, these also should manifest the same tendencies. Moreover, these analytically-constructed graphs should have some likeness to actual curves plotted from test-bed information. Unless they were thus comparable, the reasoning carried through this chapter could not be justified.

In fact, it is possible to find a good measure of agreement. The curves relating to measured pump performance, Fig. 53, have much the shape that Figs. 124 to 126 have led us to expect. In another way, their guidance has already been taken into account in Part B of the book. It will be found in § 64. The reason why a limited range of shape numbers was there recommended should now be evident. Outside that range, the probable pump efficiency has fallen so far below its maximum that the pump would not be acceptable for normal services.

As for the other main tendency, (ii), above, it has likewise been recognised in Part B of the book, viz., in § 93. In a general way, it was there suggested that the ratio $(1 - \eta_h)/(1 - \eta_m)$ increases as the shape number rises. The formulæ of § 193 permit this ratio to be put in the form $\Delta_h/(\eta_m + \Delta_h)\Delta_t$,

which in a broad sense is equivalent to the significant ratio Δ_h/Δ_t already mentioned.

A final method of representing general tendencies is to plot the actual distribution, as nearly as this can be estimated, of losses in two particular centrifugal pumps. The result is seen in Fig. 127. One of the pumps, of the low specific-speed

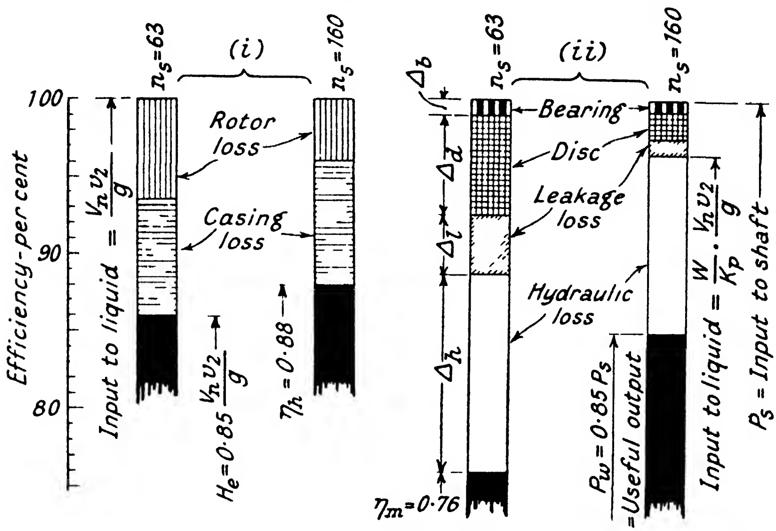


FIG. 127 Distribution of losses in low specific speed pump and in medium specific-speed pump.

diffuser type, has a shape number of 63; the other, of the medium specific-speed volute type, has a shape number of 160. On the left of the diagram, (i), energy losses and hydraulic efficiencies are given; on the right, (ii), power losses and gross efficiencies are plotted (*).

201. Power Losses in Axial-flow Pumps. Although the tendencies to be studied here are less clearly defined than they have been hitherto, we are at least entitled to say :—

Disc Friction Loss P_d. A precise answer can be given on this point. The absolute loss and the relative loss are alike *zero*, because there is no disc.

Leakage Loss P_l. Using Fig. 70 as a guide, we could imagine a range of propeller-pumps all of the same diameter, speed, and blade area, but with the blades set at various inclinations. The only difference between the conditions now in force, and

those assumed to be operating in § 104, is that it is here regarded as permissible to make minor blade-angle adjustments to ensure for each pump its maximum efficiency. As the head remains unchanged throughout, the leakage power loss will likewise be unaffected; but the input power steadily rises in sympathy with the increase in blade inclination and the increase in discharge. Thus the *relative* leakage loss will diminish as the specific speed or shape number increases.

Hydraulic Loss. Thinking in a crude way of the frictional loss as the liquid flows over the surfaces of the rotor blades, this should vary as the square of the relative velocity. But this relative velocity should not materially change as the blade pitch steepens, from which it follows that the *rotor* relative hydraulic loss will decline as the shape number rises.

On the other hand, the relative hydraulic loss in the casing and *recuperator* of a given axial-flow pump will increase as the discharge increases.

Gross Efficiency. Thus it again seems likely that the curve between shape number and gross efficiency will droop below permissible limits if the value of the shape number is not kept within a restricted range, § 64. It is pertinent in this connection to study Fig. 146, which represents the performance of what is, *in effect*, a range of pumps such as is now in question.

CHAPTER XIV

PERFORMANCE UNDER REDUCED-FLOW AND INCREASED-FLOW CONDITIONS

202. Departures from Design Conditions. Although the design of a pump is primarily controlled by certain specified values of head, speed, and discharge, it would not be reasonable to expect that those conditions could invariably be maintained throughout the working life of the pump. The user will doubtless be very glad to alter the output to suit momentary changes of demand; there may be unavoidable changes in head imposed by service conditions. Such changes may occur, for example, if the pump draws from a well, a river, or a tidal estuary. It is true that the pump speed may not vary appreciably, especially if the set includes a normal A.C. motor; yet on the other hand speed variations may be very helpful in securing desired changes in head or discharge.

In this chapter we still propose to hold the *speed* at its original designed value: only the head and the discharge of the pump will be allowed to vary. On the test-bed these variations can very easily be realised by throttling, § 172, while on the site such influences as those just mentioned may come into effect.

In studying the resulting change in the pump performance, it will be logical to observe first how the general flow picture is distorted, and in turn to see how the energy *received* by the liquid and the energy *lost* is altered. Following the nomenclature of § 159, the typical conditions now in question are

(i) reduced-flow, when the discharge is less than normal (with the special case of zero-flow) and (ii) increased-flow, when the discharge is greater than normal. It is to be noted that all such variations are fundamentally different from those examined in Chapter XIII. What was in question there were successive ranges of rotors *each designed for a specific duty*; what we are to study now is one rotor working under conditions for which it was not designed at all.

It will be helpful henceforth to use the terms *specific speed* and *shape number* in a new and special sense. Strictly speaking, any alteration to the value of head H and discharge Q will mean a change in the numerical value of N_s or n_s. These

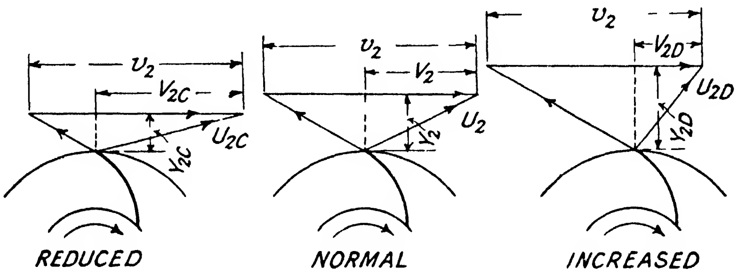


FIG. 128.—Ideal outlet velocity diagrams for reduced, normal, and increased flow.

modified values will be disregarded. When speaking of a pump working under reduced flow or increased flow conditions, we shall continue to describe it by the specific speed or shape number *applicable to its original or design performance*

203. Changes within Centrifugal Pump Impeller (*) It is not difficult to form an impression of what will happen when the discharge or rate of flow through the rotor is altered. Two factors will remain unchanged. They are: (i) the rim velocity v_2, because the shaft speed is by definition held steady, (ii) the ideal (relative) angle γ at which the liquid leaves the wheel, because the blades have suffered no change and give just the same guidance as before.

Factors that will be modified include all flow velocities. Both the radial flow component Y, and the relative velocities v_r through the wheel passages, will rise or fall in harmony with the variations in discharge Q. In consequence, the original

outlet velocity diagrams *will no longer be valid*. As the whirl component V may feel the effect of the distortion of the diagrams, a final consequence is that the *effective head* itself may be influenced.

For an ideal impeller with very numerous blades, the changes under review could be represented by the diagrams plotted in Fig. 128. In these and in subsequent diagrams, new subscripts distinguish departures from the normal regime. The subscript $_C$ will denote reduced-flow conditions, and the subscript $_D$ increased-flow conditions. Referring to Fig. 128, we observe that if the flow is *reduced*, then :—

The tangential whirl component *increases* from V_2 to V_{2C}.
The absolute outlet velocity *increases* from U_2 to U_{2C}.
The flow component *decreases* from Y_2 to Y_{2C}.

If, on the contrary, the flow is *increased*, then :—

The tangential whirl component *decreases* from V_2 to V_{2D}.
The absolute outlet velocity *decreases* from U_2 to U_{2D}.
The flow component *increases* from Y_2 to Y_{2D}.

In conformity with these variations of velocity of whirl there must also be variations of ideal energy $E_\infty = \dfrac{V_2 v_2}{g}$. Although the corresponding variations of effective head H_e may not keep pace with these changes, they will probably be of the same nature; and to this correlation the pump owes one of its most valuable attributes, viz., its *power of self-regulation*. Thus if for any reason the external head to be overcome rises above the head that the pump is momentarily generating, then automatically the discharge declines by just the right amount to enable the impeller to give the necessary increased head. Conversely a reduction in external resistance will stimulate the pump to supply more liquid and in this way to generate a lower head.

204. New Kinds of Energy Loss. An examination of the distorted *inlet* velocity diagrams, Fig. 129, shows that additional energy losses are now to be reckoned with. If the blade tips were made tangential to the *designed* inlet relative velocity v_{r1}, they cannot at the same time be tangential to the *modified* velocities v_{r1C} or v_{r1D}. Nor will energy dissipation be quickened by this cause alone. When the rate of flow is below normal,

there will be a tendency for the liquid stream in each of the impeller passages to concentrate near the front of the blades, leaving a more or less dead space near the back of the blades: dead in regard to effective forward motion, but alive in its

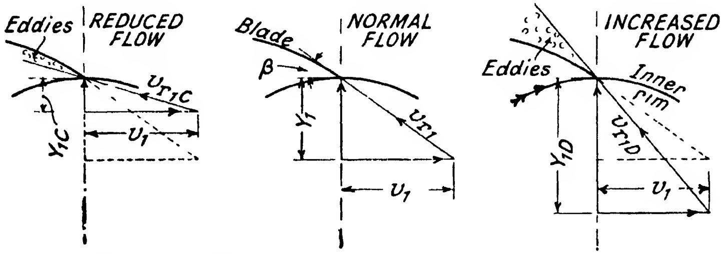


FIG. 129.—Normal and distorted inlet velocity diagrams.

capacity to waste energy. This eddying effect has already been suggested in Fig. 16 (iii).

In the recuperator also the flow conditions deteriorate. When the pump discharge is reduced, there will inevitably be heavier energy losses in the volute. While the velocity of the liquid leaving the impeller and entering the volute is *above* normal, as Fig. 128 shows, yet the mean velocity in the volute itself must be *below* normal. Thus the essential conditions for maximum efficiency of energy conversion, § 45, can no longer apply. If the pump discharge is considerably above normal, then the absolute outlet velocity vector, Fig. 128 (iii), takes on an abnormal inclination. Very serious contraction of the main stream of liquid may occur at the volute tongue, Fig. 130 (i), with consequent additional eddy loss.

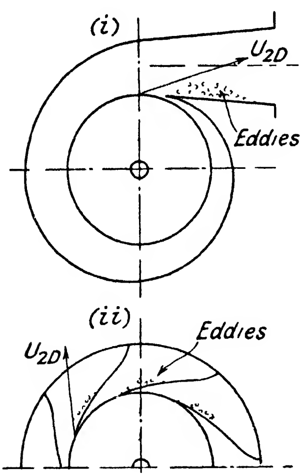


FIG. 130.—Eddy losses in recuperators.

In regard to guide-blade or diffuser types of recuperator, we observe from Fig. 130 (ii) that departures from designed rates of flow will again destroy the necessary correspondence between blade inclination and velocity vector inclination. Energy dissipation is set up here also.

205. Changes in Efficiency. In trying to assess the overall effect of flow variations on the gross efficiency of the pump, we can find guidance in a paragraph—§ 198—that has already explained some of the consequences of changes in discharge. As a first approximation it is still true to say that the "non-hydraulic" losses P_b, P_d, P_l remain virtually unchanged; but the law of variation of the hydraulic loss P_h will certainly be radically modified by the additional eddy losses just examined, § 204. Their immediate effect will be to augment the items h_t, h_t, which in Fig. 121 denoted energy wastage at inlet to rotor and to recuperator. Their ultimate effect must be to lower in turn the hydraulic and the overall efficiency of the pump.

In brief, the pump can only be expected to give its best performance at its design point. If the flow is greater or less than the designed flow, *the efficiency of the pump will inevitably suffer.*

206. Zero-discharge Conditions. Only when the discharge falls so low that it approaches its limiting value of zero do we find existing methods of analysis inadequate. If no liquid at all issues from the delivery flange, graphs such as Fig. 121 lose much of their significance. Since the discharge W is zero, then the output power P_u is zero and the hydraulic loss P_h is apparently zero as well, no matter what may be the values of the individual items h_{lr}, h_{ld}, etc. Thus we should expect the input power P_s to be equal to the sum of the mechanical loss P_b, the disc friction loss P_d, and the leakage loss P_l, all of which are assumed to have undergone no substantial change. But in fact the measured input P_s is *much greater* than this. The observed zero-discharge value is hardly ever less than about one-third of the normal or designed power input, and it may be considerably more. *Some additional type of energy dissipation* is evidently taking place inside the pump, that cannot be represented by diagrams such as Figs. 121 and 127. One may describe it as a very violent and confused churning effect at the inlet and outlet zones of the rotor. The liquid in the rotating impeller passages might be imagined to be "scrubbing" against the *nominally* stationary liquid in the inlet and outlet passages of the casing. We can easily believe that in these casing passages the liquid is in truth very far from being at rest,

because we know that so rapid a rate of energy loss could only be accounted for by unusually fierce turbulence propagated throughout a considerable mass of liquid.

When the pump is running against closed-throttle, then, we steadily feed into it energy at a rate P_s, but no energy seems to be coming out of the pump. It does not take long to find out what has happened to the energy input, though. It has all been converted into *heat energy*. After a few minutes the liquid grows hotter and hotter, and this temperature-rise will continue until either (*a*) the whole assembly attains a temperature at which the heat radiated to the surrounding air just balances the energy input, or (*b*) the liquid boils or the shaft seizes or some other violent interruption occurs. In ordinary installations it is the attendant's duty to prevent such happenings, but in high-power pumps it may be necessary to provide protective systems including automatic leak-off passages.

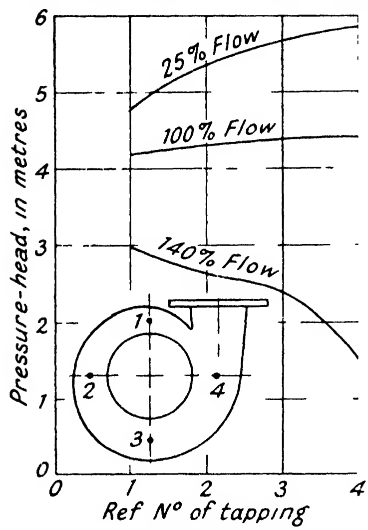


FIG 131 Normal and abnormal pressure distribution in volute

207. Other Consequences of Discharge Variation. (i) *Side Thrust on Impeller.* The possibility of uneven pressure-distribution around the impeller rim has already been mentioned in § 73 (*c*). Here it is to be noted that even if the unbalanced side thrust shown in Fig. 41 is absent under design conditions, it may become quite severe under reduced-flow conditions. Figures from an actual volute pump are plotted in Fig. 131; they show that at 25 per cent. flow (reduced-flow), the pressure-head in the volute rises steadily as one follows the gauge-points in the direction of rotation of the shaft. At normal or 100 per cent. flow, the pressure is nearly uniform, as it should be. At increased-flow, the pressure-changes are opposite to what they were at 25 per cent. flow.

(ii) *Abnormal Velocity Distribution at Outlet Flange.* The

question of irregular velocity-distribution at the pump outlet flange was studied in § 160. When the pump is working at partial flow such irregularity may be much intensified. Although the experimental observations depicted in Fig. 132 relate to a screw pump, rather than to a radial-flow centrifugal pump, they have so many points of interest that they may fitly be included here. First of all the directions of the velocity vectors U_3 conclusively prove the existence of the secondary circulation within the volute that was described in § 101 ; this type of motion may occur in all offset or non-symmetrical

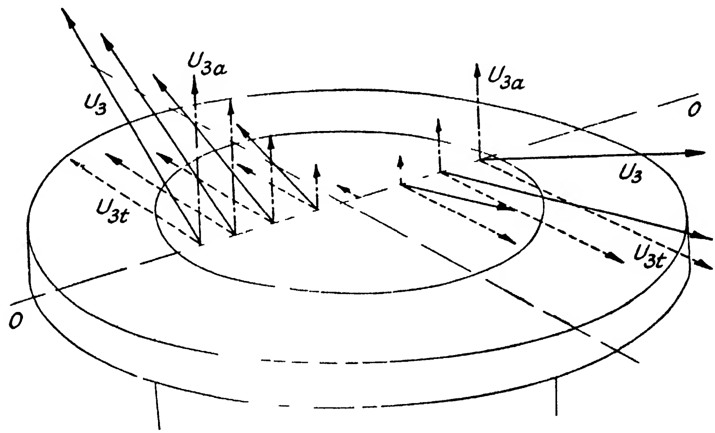


FIG. 132. Abnormal velocity distribution at outlet flange of screw pump, working at 30 per cent. flow.

volutes such as those shown in Figs. 51 (iii), 64, and 65. It is analogous to the reverse-flow that may arise in axial-flow pumps, § 209. Then the vectors show, Fig. 132, that at part-flow the components U_{3t} in the plane of the delivery flange may be much more powerful than the components U_{3a} in the direction of the delivery pipe axis. Not only does this fact throw additional doubt upon the nominal figures which purport to represent total energy at the outlet flange, § 160, but it explains why these disturbances of flow may be propagated so far down the delivery pipe itself as to falsify quite seriously the readings of the meter which records the pump discharge, e.g., as in Fig. 116 (iv).

(iii) *Noise, Cavitation, etc.* If the flow abnormalities at partial discharge are sufficiently pronounced, we may expect

more direct evidence than is shown by instrument readings. The pump may vibrate, it may run noisily, and if the conditions of reduced-flow are maintained for a long time there may be rapid erosion of the rotor or recuperator blades and passages, § 257.

208. Changes in Axial-flow Pump. Pumps intended for three-dimensional flow have the peculiarity that even under design conditions a single outlet velocity diagram no longer serves to give a true flow-picture. There must be individual diagrams for each directive surface; two such diagrams were plotted in Fig. 23. The question we now have to ask is: when the rate of discharge is raised or lowered, will all these different diagrams be affected equally, or will they suffer varying degrees of distortion? An axial-flow rotor will be the easiest to study, and we may conveniently think of the directive surfaces, § 29, as being actual metallic boundaries rather than mere abstractions. Instead of having one rotor, that is to say, we have a number of cylindrical rotor-rings fitting one within the other, Fig. 133. Here are drawn ideal outlet triangles, (i), for full-flow and, (ii), for part flow, relating both to an outer rotor ring and an inner rotor-ring. At once it is evident that the *percentage* increase in tangential velocity component V_2 is much greater for the outer ring than it is for the inner ring, from which it follows that at part-flow the outer ring may be expected to *generate a higher head* than the inner ring.

But this conclusion cannot be admitted. If a uniform pressure prevails across the inlet plane to the pump and also across the outlet plane, there must necessarily be a *uniform* energy-increment in all the rotor-rings. This difficulty is readily overcome. It only arose because of an unjustifiable assumption that the reduction in discharge applied uniformly to all the rotor-rings. What is much more likely to happen is that the self-regulating process described in § 203 will have a differential effect, automatically enabling each ring to create the same head but allowing less liquid to pass through the inner rings (in relation to normal conditions) than passes through the outer rings. The general state of flow could then correctly be described by the diagrams in Fig. 133 (iii). In any event, a reduction in discharge is accompanied by a proportionally *greater* increase of head than was found in a centrifugal pump.

209. Back-flow and its Consequences. As the main flow through the pump progressively declines, consequent upon a still further increase of the effective head, there will come a stage at which the flow velocity in the innermost rotor-ring sinks to zero. What will happen if the discharge is throttled down still more? Manifestly the direction of flow in this inner

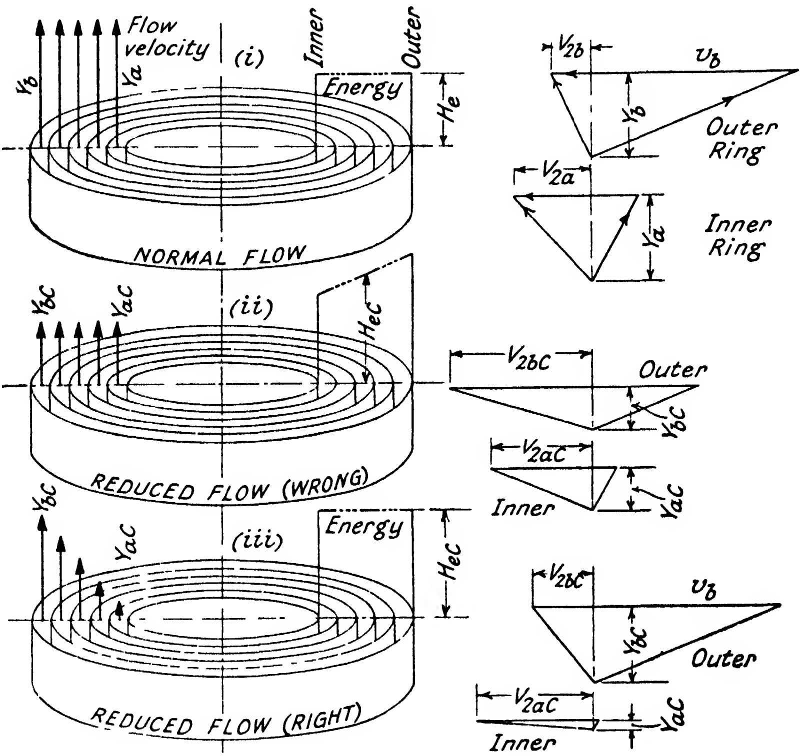


FIG. 133.—Effect of reduced-flow conditions on ideal axial-flow rotor velocity triangles.

ring will be reversed: liquid will begin to *run backwards* through this part of the rotor, Fig. 134 (i). When the final stage equivalent to closed-throttle is reached, quite a brisk circulation of liquid will be in progress, forwards through the outer rings and backwards through the inner rings. The general character of the flow will not be destroyed if we now remove the annular partitions as having served their purpose, leaving us with the type of secondary motion suggested in Fig. 134 (ii).

In regard to the fundamental question of power consumption we cannot now pretend that the hydraulic power loss P_h is even apparently zero. Energy will be required to maintain the flow through the outer part of the rotor passages, nor will this energy be returned to the blades when the liquid returns through the inner part of the passages. On the contrary, the blades must give still more energy to the liquid. Naturally the net benefit we receive from this expenditure of energy is *nil*, because no liquid is coming out of the pump delivery branch.

210. General Trends in Axial-flow and Mixed-flow Pumps. The power loss associated with back-flow will certainly be an important item in the total closed-throttle input P_s of an axial-flow pump. We have also to remember that at low rates of discharge the shock losses, h_t, h_t, Fig. 121, at entry to rotor and recuperator blades are relatively much more serious than in a centrifugal pump, § 204. This is because the changes in inclination of the rotor inlet relative velocity vector v_{r1}, and of the outlet absolute vector U_2, are now more pronounced, as may be realised by comparing Fig. 133 (iii) with Figs. 128 to 130. Without exception, then, the diverse effects of reducing the flow through the pump all tend in the same direction– and that is in the direction of intensifying the rate of energy dissipation. This is shown on the test-bed by a steady *rise* of power input as the throttle-valve is progressively closed, terminating at zero discharge by a value P_s which is a good deal *greater* than the normal power corresponding to design conditions.

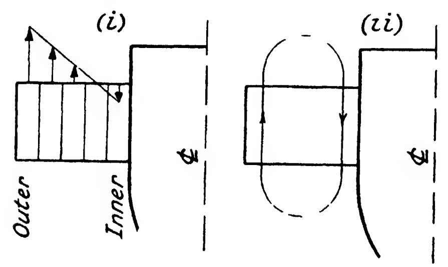


FIG. 134.—Reversed flow in axial flow rotor.

The remedy for this decline in efficiency is to use a *variable-pitch propeller*, § 104. If the blade inclination itself can be adjusted to agree more or less with the inclination of the velocity vectors irrespective of the rate of discharge, then one of the major sources of power loss is eliminated.

As in nearly all other respects, *mixed-flow pumps* exhibit traits under abnormal flow conditions that are intermediate between those of radial-flow pumps and those of axial-flow

pumps. Thus the closed-throttle power input P_s for a screw-type pump is very nearly the same as the normal input; it is higher than for an equivalent centrifugal pump, but lower than in an equivalent (non-adjustable) propeller pump. What has a good deal to do with the more economical part-flow performance of the screw-type pump is the ratio d_{2a}/d_{2b} of the rotor diameters, § 106. The more nearly this ratio approaches unity, the more nearly is there uniformity across the outlet area of the rotor; and we have already seen that it is extreme lack of uniformity that promotes back-flow with corresponding power wastage in an axial-flow pump, § 209.

Characteristic Curves at Constant Speed

211. Graphical Plotting of Pump Performance. Just as the variations studied in Chapter XIII could be recorded in graphs such as Fig. 125, so we ought to be able to find a convenient way of plotting the variations in the regime of a particular pump. The obvious system is found to be the best—to plot discharge as abscissæ, and head, power input and efficiency as ordinates. The resulting *characteristic curves* are of great practical utility in giving to the designer and the user a clear picture of what the pump will do in specified circumstances. They are not, like earlier performance graphs, based on analysis or speculation. They are the actual records of test-bed observations or at least they are closely linked with such figures (*).

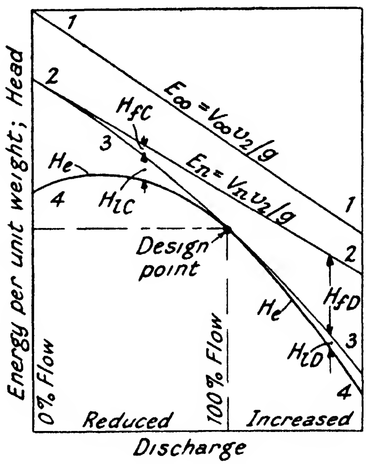


FIG. 135. Relation between rate of discharge, and energy, in centrifugal pump.

Fig. 135 serves as a summary of the reasoning carried through §§ 203 to 206. It will give a preliminary impression of what a *head-discharge* characteristic curve will look like. We begin with the uppermost line (1-1) which shows how the ideal energy input per unit weight of liquid, $E_\infty = \frac{V_\infty v_2}{g}$, depends

upon the discharge ; in conformity with what was said in § 203, the energy continuously declines as the quantity increases. Next comes the line (2-2) suggesting the corresponding variations in the true input E_n ; the intercept between these two lines gives a measure of the effect as the number of impeller blades falls from ∞ to n, §§ 17, 24. To obtain the value of the effective head H_e, we must now deduct the total hydraulic energy loss H_L. In Fig. 121 this total amount was analysed into its constituent elements, but here we can more usefully collect these into a group of friction losses, H_f, and a group of eddy, shock, or separation losses H_l. The friction losses, being dependent upon the rate of flow of liquid through the pump passages, will mount from a value such as H_{fC} at reduced-flow to a value H_{fD} at increased-flow, Fig. 135. In accordance with §§ 204, 205, the shock or eddy losses are assumed to have the value zero at the design point, and they are represented by H_{lC} or H_{lD} at other flow rates. Successive subtractions thus yield in turn the line (3-3), and finally the desired *head-discharge characteristic* (4-4). Here at last is the curve which shows the inter-relation between discharge and effective head, for a centrifugal pump running at constant speed.

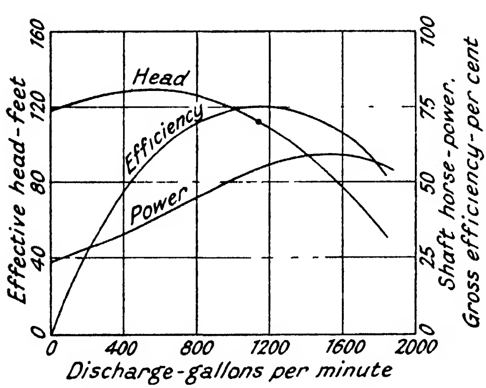


FIG. 136 Centrifugal pump characteristics. $N = 900$ r.p.m., $n_s = 60$.

212. Head, Power and Efficiency Characteristics.

Turning now to actual records derived from the test-bed, they are usually put in the form shown in Fig. 136. With the pump speed maintained steady, the flow is progressively throttled, § 171, and successive readings of head, discharge, and power input are taken. The complete performance is finally plotted as indicated.

The *head-discharge characteristic* has just the shape that the

reasoning of the previous paragraph, as embodied in Fig. 135, has led us to expect. The *power-discharge characteristic* clearly shows the importance of the zero-discharge or shut-off loss predicted in § 206. The decline of the *discharge-efficiency characteristic* on either side of the design point is in line with the arguments of § 205. What is especially to be remembered is the remarkable tenacity and firmness with which each curve maintains its appointed shape. Although the term *rotodynamic* is used to exclude all types of *positive* pumps, yet nothing could be more positive than the rotodynamic pump's adherence to a characteristic kind of performance. Nothing can alter it. So long as the pump is working properly and is kept in good condition, it will insist on discharging a particular quantity of liquid if a particular head is maintained; and similarly if we want the pump to give a predetermined discharge we have no other method of control than to modify the head accordingly. Of course if we could vary the speed as well that would be quite another matter, but in this chapter we have agreed to maintain a *constant* speed. All the characteristic curves are plotted on that basis. **(Example 22.)**

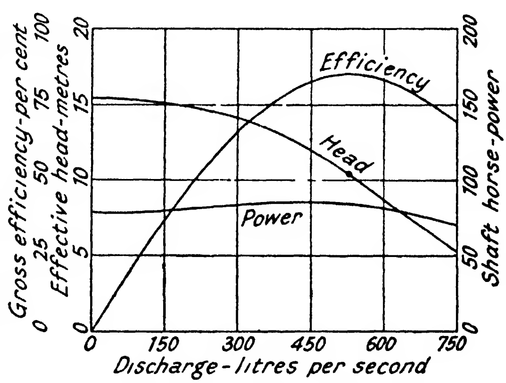


FIG. 137. Screw type pump characteristics. $N = 730$ r.p.m., $n_s = 280$.

213. Rotor Shape and Characteristic Shape. The primary advantage of the conception of shape number is that one number only serves as a kind of shorthand description of the geometrical *proportions* of the rotor, § 61. In a similar way the characteristic curves reflect the *behaviour* of the rotor (*) As the shape number changes, so should the form of the characteristics alter. Examples of this inter-dependence are offered in Figs. 136 to 138, which describe in turn the behaviour at constant speed of a centrifugal pump, a screw-type pump and an

axial-flow pump. Because of the variety of units employed—a diversity that cannot be escaped in practice—the information is

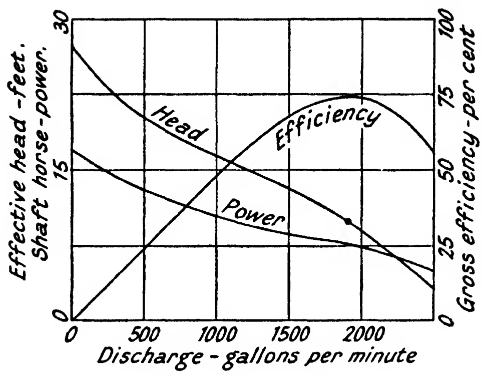
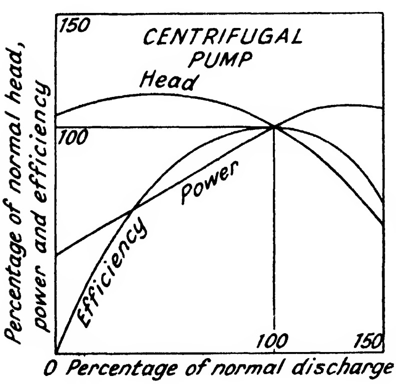


FIG. 138. - Propeller pump characteristics. N = 1300 r.p.m.; n_s = 640.

not given in the best form for comparative purposes ; so once again we must devise some non-dimensional system. Here it



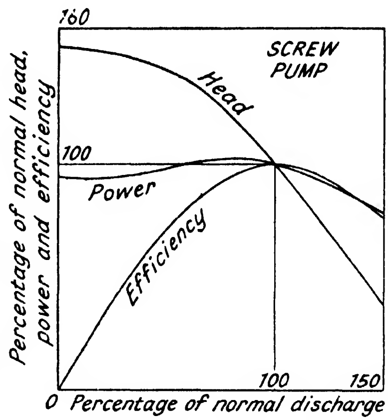


FIG. 139. (from Fig. 136). FIG. 140. (from Fig. 137).

Plotting of pump characteristics on percentage or comparative basis.

will be instructive to express all the variables as a percentage of their respective values *at the design point.* This method will have the additional convenience of permitting a numerical assessment of the terms reduced-flow and increased-flow. A percentage flow of 56 will mean that the discharge is 56 per cent.

of normal or designed flow; a percentage flow of 145 will signify a fairly extreme state of increased-flow.

In such a manner, then, the graphs of Figs. 136 to 138 can be transposed into those of Figs. 139 to 141, thereby yielding a condensed and informative statement of what §§ 203 to 210 have already given in general terms. It is natural to expect that for pumps having shape number intermediate between those chosen here, the characteristic curves would also have an intermediate character. If, then, we were to examine a full range of characteristics, we might reasonably predict from Figs. 139 to 141 that they would reveal the following tendencies:—

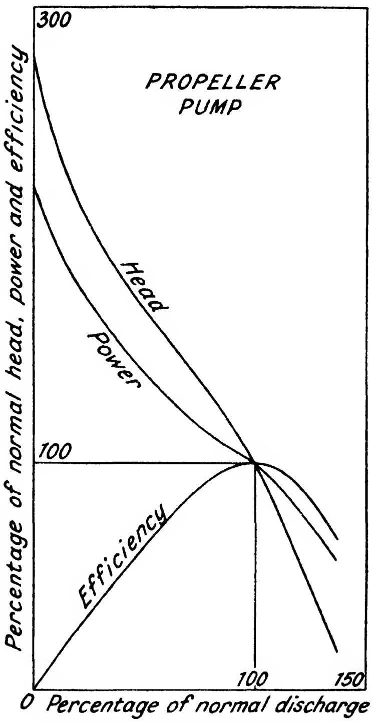


Fig. 141. (from Fig. 138).

(i) The shut-off or closed-throttle *head* would rise in sympathy with an increase of shape number.

(ii) The closed-throttle *power* would likewise become greater as the shape number became greater.

(iii) The summit of the *efficiency* curve grows progressively more peaked or less rounded as the shape number rises.

Throughout these discussions, it is clearly to be understood that the term *shape number* is used in the special sense defined in § 202.

214. Control of Characteristic Shape. For general purposes of design, it was helpful to assume that a particular shape number corresponded with an established set of ratios ϕ, ψ, and λ, Fig. 54. But there is no finality in such arbitrary groupings; we have seen in §§ 196 to 199 that individual values

of the variables can be manipulated independently. What will be the effect of these changes on the shape of the pump characteristics? It is important to know, because the operating conditions to which rotodynamic pumps must conform often require a special shape of characteristic. Such curves as those in Figs. 139 to 141 are therefore to be regarded as generic only; they form a valuable companion picture to such as diagrams Fig. 55.

Beginning with a centrifugal pump of low specific speed, we

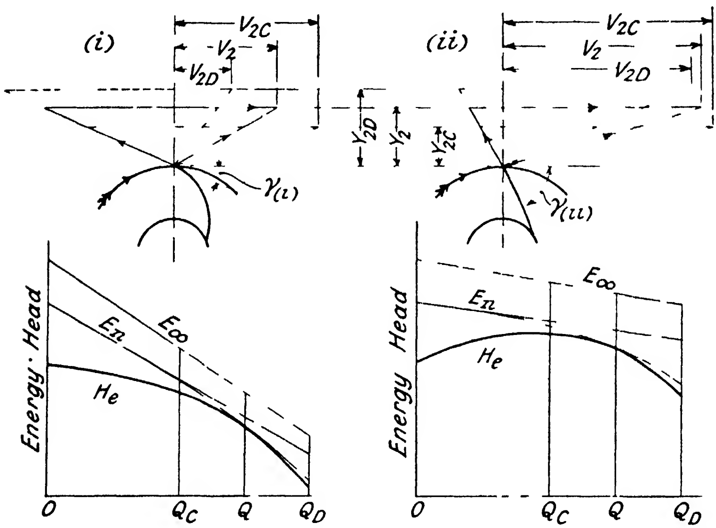


FIG. 142. Effect of blade angle on shape of head discharge characteristic

may first study the effect of variation of the outlet blade angle γ; N and v_2 remaining invariable. In Fig. 142 there are shown ideal velocity diagrams for various percentage flows, relating to an impeller with a small angle, (i), and with a very large angle, (ii). In the one case variations of flow velocity produce very considerable changes in the whirl component V_2; in the other case the effect is relatively trifling. But flow velocity is proportional to discharge, and for present purposes whirl velocity is proportional to ideal energy E_∞, § 203; thus we can quickly construct the ideal discharge-energy graphs as in Fig. 135. By making such reasonable deductions as went to the tracing of curves (1-1), (2-2), in that diagram, we can predict with some

confidence that the corresponding head-discharge curves will have some such forms as are sketched in Fig. 142. There we see in very legible characters the influence of change of blade angle: the *smaller* the angle, the more quickly does the head decline as the percentage flow increases.

A steeply-falling head-discharge characteristic as in Fig. 142 (i) is also to be had by other means. Let the outlet blade angle be increased from $\gamma_{(i)}$ to γ', but let the flow velocity be increased to the same extent. That will give the range of velocity diarams drawn in Fig. 143. The tendencies there revealed are identical with those of Fig. 142 (i), and thus we may expect the same consequences. Now there is a direct link between flow velocity and flow ratio ψ; and also between flow velocity, blade angle, and speed ratio ϕ. As the width ratio λ can be adjusted independently of these, and as specific speed or shape number can be expressed in the form $\phi\sqrt{\lambda\psi}$, § 59, we can admit that it *does* lie within our power to control the shape of the head-discharge characteristic without altering the specific speed.

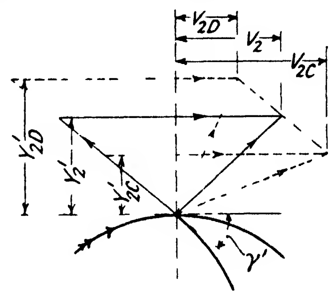


FIG. 143 —Alternative method of controlling characteristic shape.

215. Terminology of Characteristics.

In order to identify the two distinct shapes shown in Figs. 142, 144, they are given the names: –

(i) *Stable head-discharge characteristic.*

(ii) *Unstable head-discharge characteristic.*

In a stable curve there is only one rate of discharge corresponding to a specified head (except perhaps at very small percentage flows), while if the curve is unstable there may be two. Thus according to the unstable curve seen in Fig. 136, a head of 120 ft. can be associated either with a discharge of 60 g.p.m., or a discharge of 950 g.p.m.; nor is it possible to be sure which of these the pump may choose to deliver. In some kinds of installation this uncertainty may have such dangerous consequences that pumps with unstable characteristics cannot be accepted at all, § 346 (ii).

Allied with the two classes of head-discharge curve there

are two corresponding types of power-discharge characteristic. They are illustrated in Fig. 144 and they are entitled :—

(iii) *Non-overloading* curve which is related to the stable head-discharge curve.

(iv) *Overloading* power curve, related to the unstable head-discharge curve.

The reason for the diversity of shape is fairly clear. Under corresponding increased-flow conditions, for instance, the stable curve indicates a lower head than the unstable curve, but there is nothing to suggest that there is any substantial difference in efficiency. The equivalent power curve should therefore also be lower, which is indeed what we find. As for the terminology,

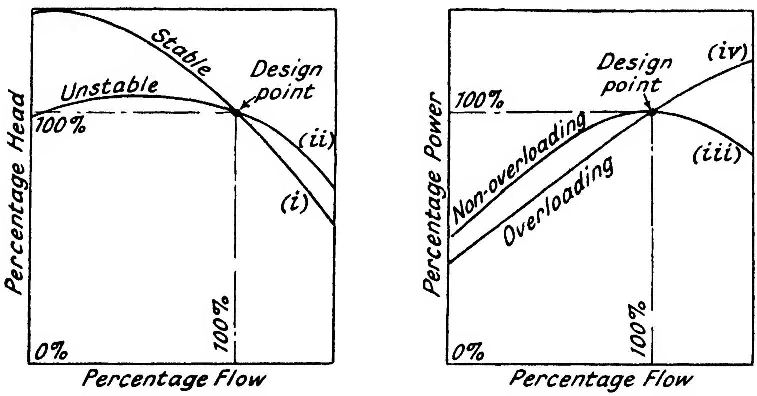


FIG. 144.—Contrasting types of head- and power-characteristics.

" non-overloading " suggests that the motive unit driving the pump will never have to yield more than a known maximum output ; it cannot be overloaded because the curve rises to a limiting height, Fig. 144 (iii). But the " overloading " curve continues to rise indefinitely.

216. Other Methods of Controlling the Characteristic Shape. (*a*) *In Centrifugal Pumps.* (i) The diameter ratio d_1/d_2, § 92 may have a slight effect on the run of the head-discharge curves, especially near the zero-discharge point. (ii) Comparing two pumps having identical rotors, the machine with *volute-type* recuperator might have characteristics slightly different from the machine with a *diffuser-type* recuperator, §§ 44, 45. (iii) In multi-stage pumps there is the possibility of using in successive stages different impellers and different

recuperators. Thus some of the stages might have volutes and some diffuser-rings (*). The overall effective head generated by the pump at various rates of flow will thereby follow a characteristic intermediate between the individual characteristics of the separate stages.

(*b*) *In Propeller Pumps.* (i) The presence or absence of inlet guide-blades, Fig. 68, has a marked effect at low percentage flows. Without the guides there is no doubt that pre-rotation occurs—the liquid before reaching the rotor has already acquired a tangential velocity component, which prevents the rotor blades from impressing on the liquid the head that we should expect. The fixed inlet blades discourage this motion, and the effective head is consequently higher, Fig. 145. (ii) The advantage of the *variable-pitch propeller,* § 104, is very well shown in Fig. 146. Considering the blades of a pump so fitted to be temporarily set in turn at each of six different positions, I, II, III, etc., then for each position a head discharge and an efficiency-discharge characteristic could be obtained. exactly comparable with those plotted in Fig. 138. These are seen in Fig. 146. But at the constant working head—in this instance 5 metres—the pump is manifestly running nearly at the peak of its efficiency. In practice, the desired discharge is obtained by hand regulation of the rotor blades; and at whatever percentage flow is selected, the pump efficiency will approach the highest value that a machine of this general type can possibly yield.

UNIVERSAL FLOW CONDITIONS

217. Total Range of Pump Performance. Having observed how changes in rotor shape or changes of head or discharge can influence the behaviour of a pump running at its *designed* speed, we can proceed to study the effects of changes of :—

(i) Speed of pump.
(ii) Size of pump.
(iii) Liquid pumped.

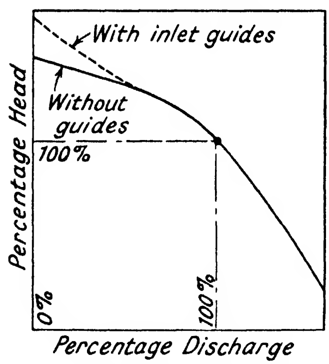


FIG. 145.—Effect of inlet guide-blades. (§ 216.)

The term " Universal Flow Conditions " implies, then, that we must be prepared to deal with pumps of any size or shape, running at any speed, delivering any kind of liquid at any rate of discharge against any head—subject, of course, to reasonable limits.

PUMP BEHAVIOUR AT VARYING SPEED

218. Equivalent Design Conditions. Speed variations have already been examined in an elementary fashion in § 53.

That paragraph showed that if the head and discharge were suitably adjusted, the speed could be altered *without altering the efficiency.* It was essential to regulate the variations so that the speed ratio ϕ and the flow ratio ψ themselves remained fixed, from which it followed that the variables were linked by these relationships :—

$$\text{Discharge} \propto \text{Speed},$$
$$\text{Head} \propto (\text{Speed})^2,$$
$$\text{Power} \propto (\text{Speed})^3.$$

If an actual pump is running at a speed below its designed speed, and the discharge is controlled so that these requirements are as nearly as possible fulfilled, then it is convenient to say that the pump is working under *equivalent design conditions.* When

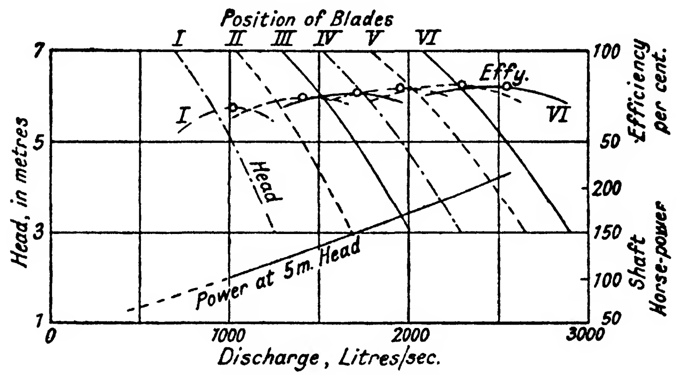


FIG. 146.—Characteristics of variable-pitch propeller-pump, at constant speed. (§ 216.)

" design conditions " have been established, we understand that the pump efficiency is the maximum attainable at the designed speed N_d. Similarly when the pump is running at any lower speed N_{ed}, the highest efficiency for that reduced speed will only prevail if " equivalent design conditions " have been fulfilled. According to the idealised treatment of § 53, these two values of maximum efficiency were identical ; but test-bed results show that in fact they are not equal. The discrepancy can be understood by carrying through an analysis of relative power losses on the lines of §§ 186 to 194. As this departure from ideal conditions may make it impossible to achieve simultaneously the three basic relationships just enunciated, we may

henceforth stipulate that the discharge will accurately be regulated in direct proportion to the speed, the head and power being left to adjust themselves as well as they can. Thus we can be sure that all *velocities*, whether of the rotor or of the liquid, will also vary *directly as the speed.*

219. Influence of Pump Speed upon Energy Losses. On the understanding, then, that equivalent design conditions prevail, viz., that $Q \propto N$, the analysis can proceed thus :—

(i) *Mechanical Loss.* In a given pump, the correlation between speed N and mechanical power loss P_b, § 188, will probably be of the form $P_b = k_1 . N^m$, where the value of the exponent m may range from about 1 to 2, depending upon the type of bearings and stuffing-boxes. Although we are no longer sure that the input power P_s will vary exactly as the cube of the speed, we shall find later on that the departure is not serious, and thus for present purposes we may still write provisionally :— $P_s = k_2 . N^3$. Thus the relative mechanical loss P_b/P_s may be written :—

$$\triangle_b = \frac{k_1 N^m}{k_2 . N^3}.$$

The value of this ratio evidently *increases* as the speed drops. (The symbols k_1, k_2 . . . denote constants *applying to a particular pump.*)

(ii) *Disc Friction Loss.* Accepting formula (13 — 3), § 190, as a permissible expression for the disc friction power loss, then for a *given* pump using a *given* liquid the relation between speed and power loss can be simplified thus :—

$$P_d = k_3 . N^{2.85}$$

It follows that the relative disc friction loss

$$\triangle_d = \frac{P_d}{P_s} = \frac{k_3 . N^{2.85}}{k_2 . N^3}$$

also *increases* as the speed decreases.

(iii) *Leakage Loss.* According to §§ 83 and 192, the rate of flow of the leakage liquid in a given pump may provisionally be expressed :—

$$q_l = C_d . k_4 . \sqrt{H}.$$

Accepting tentatively the ideal law $H \propto N^2$, we find that the leakage power loss P_l may be written :—

$$P_l = C_d \, . \, k_5 \, . \, H\sqrt{H} = C_d \, . \, k_6 \, . \, N^3.$$

Thus the value of the relative leakage loss is :—

$$\triangle_l = \frac{C_d \, . \, k_6 \, . \, N^3}{k_2 \, . \, N^3}.$$

Apart from variations in the coefficient C_d, then, the relative leakage loss appears to be *independent* of the pump speed. But it is improbable that this coefficient will in fact remain unchanged. On the contrary, experience with small circular orifices indicates that the coefficient will increase as the flow diminishes. If this is true also for annular clearance spaces—and there is strong evidence to support this belief—then it seems likely that the relative leakage power loss will *increase* as the pump speed falls.

(iv) *Hydraulic Power Loss.* As the liquid flows at a rate Q through the pump passages, it undergoes a total energy loss per unit weight of H_L, of which h_f represents friction losses and h_l represents eddy losses, § 194. If each passage behaved as a smooth circular pipe does, then the relationship between friction loss and discharge would be about : $h_f = k_7 \, . \, Q^{1 \cdot 8}$. The eddy loss on the other hand would always be represented by

$$h_l = k_8 \, . \, Q^2.$$

These equations indicate that in the actual pump the conditions may fairly be described in some such way as this :—

$$H_L = k_9 \, . \, Q^{1 \cdot 95}.$$

From § 195, it follows that the hydraulic power loss can be written : $P_h = k_{10} \, . \, Q^{2 \cdot 95} = k_{11} \, . \, N^{2 \cdot 95}$. Comparing this value with the value of the input power $P_s = k_2 \, . \, N^3$, we note that

$$\triangle_h = \frac{k_{11} N^{2 \cdot 95}}{k_2 N^3}$$

Here also, then, the value of the relative power loss *rises* as the pump speed falls.

220. Relation between Speed and Efficiency. It will now be manifest why lowering the speed of the pump will also lower its efficiency. Each of the terms $\triangle_b$, $\triangle_d$, $\triangle_l$, and $\triangle_h$ tends to rise as the speed falls, and therefore the value of the gross efficiency $\eta_m = 1 - \triangle_b - \triangle_d - \triangle_l - \triangle_h$, § 187, must inevitably diminish. We can also form an impression of the *rate* at which the efficiency declines. If we admit that the three

chief relative power losses are all of the form k_{12}/N^x, where x is a fractional exponent having the value 0·1 to 0·2, it follows that the losses are at first little affected by speed changes; only when the speed has fallen to perhaps half its designed value or less is the corresponding efficiency likely to be seriously impaired. This prediction is fully confirmed on the test-bed. In Fig. 147 (ii) a typical graph is plotted between observed percentage speed, and observed percentage efficiency, for a pump running always under equivalent design conditions. The reasoning of §§ 219, 225 will suggest that for large pumps the upper or working part of the curve may be flatter than the diagram shows while for very small pumps it may be steeper.

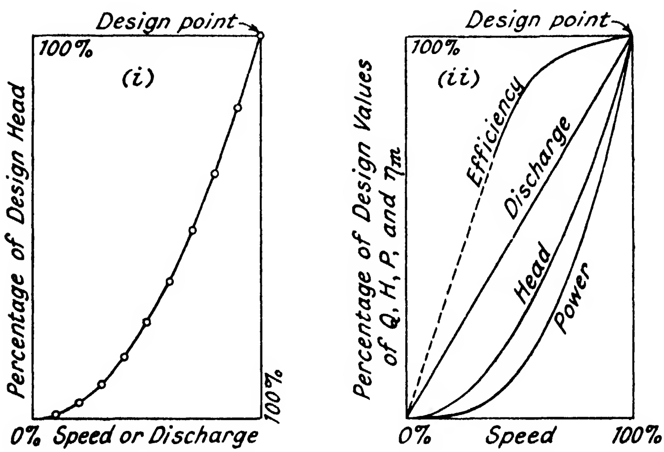


FIG. 147. - Pump performance under "equivalent design conditions".

The various graphs in Fig. 147 serve as a complete statement of how a pump responds to equivalent design conditions. At (i) there is shown the (nearly) parabolic relationship between speed or discharge, and head. Since at low speeds the relative hydraulic power loss may be slightly greater than it is at high speeds, § 219 (iv), it seems probable that the head will increase slightly faster than the law $H \propto N^2$ indicates. The individual points in Fig. 147 (i) relate to speed increments of 10 per cent. of full speed; they show how very sensitive to speed changes the head becomes as design conditions are approached. This curve is reproduced again in diagram (ii), so that the respective

variations of head, discharge, shaft horse-power and efficiency may be compared. By this time, it should be clear that the speed : power curve climbs slightly more slowly than the ideal law P $\propto N^3$ requires.

221. Characteristic Curves at Reduced Speed. Let the pump continue to run at a steady speed less than the designed speed N_d. But now we no longer intend to be bound by " equivalent design conditions ". Instead of restricting the discharge to the one value that these conditions dictate, we are going to regulate the delivery throttle-valve so as to give in turn a series of different discharges, allowing the head to

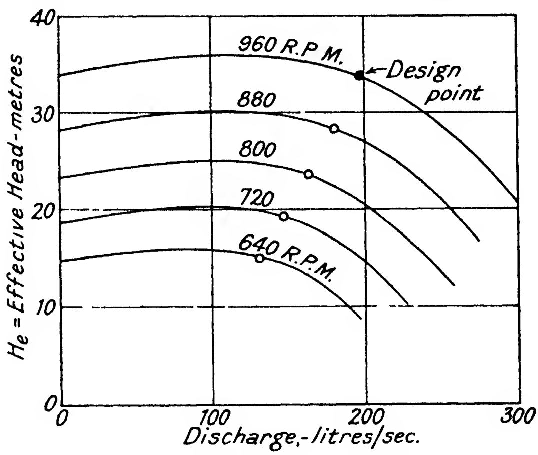


FIG. 148.– Family of head-discharge characteristics for centrifugal pump.

conform as it pleases. The resulting observations will yield characteristic curves similar in general shape to the original characteristics which served to describe " design conditions ", §§ 212, 213. Curves for other speeds can in turn be plotted, forming groups or families of graphs such as are reproduced in Figs. 148, 149, 150. **(Example 23)**

The head-discharge characteristics in Fig. 148 relate to a centrifugal pump whose design conditions are :—

Speed = 960 r.p.m.
Head = 34 m.
Discharge = 195 lit./sec.

The reduced speeds are respectively 880, 800, 720, and 640 r.p.m. Just as in Fig. 147, the " design point " and the " equivalent

design points " lie on a (nearly) parabolic curve. Since these particular head-discharge characteristics are clearly " unstable " ones, it is not surprising to find that the corresponding power-discharge curves, Fig. 149, are of the " overloading " type, § 215. In regard to the discharge-efficiency characteristics, Fig. 150, it is to be noted that a curve passing through the maximum-efficiency point of each graph would have the same general form as the typical curve plotted in Fig. 147 (ii).

222. Iso-efficiency Curves. Taken in conjunction, the three sets of characteristics shown in Figs. 148 to 150 quite effectively describe " universal flow conditions " for a particular

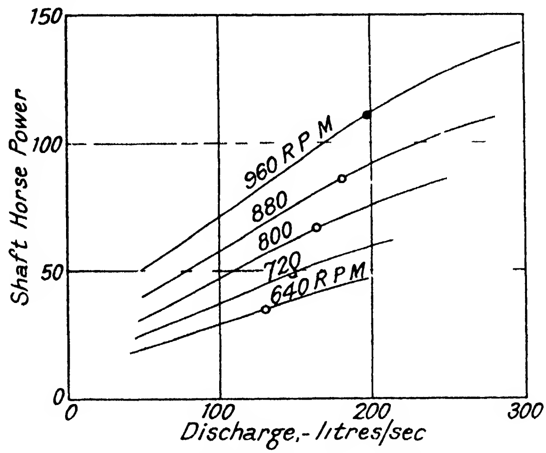


FIG. 149.—Power discharge characteristics, corresponding with Fig. 148.

pump working with a particular liquid, viz. water. They tell us what speed and power would be required to force any desired quantity of liquid against any stipulated head, within the capacity of the pump **(Example 24)**. By the use of *iso-efficiency* curves the same information can be presented much more compactly: one diagram serves instead of three. In principle a three-dimensional construction is involved. Two horizontal axes serve as scales of reference for discharge and head respectively, Fig. 151, while the third or vertical axis serves for efficiency. On the horizontal plane, normal head-discharge characteristics are plotted exactly as in Fig. 148, and from appropriate points on these curves, ordinates representing efficiency are erected. If the resulting efficiency curves were

projected on to a vertical plane, they would form a diagram comparable with Fig. 150. But what we now wish to study is

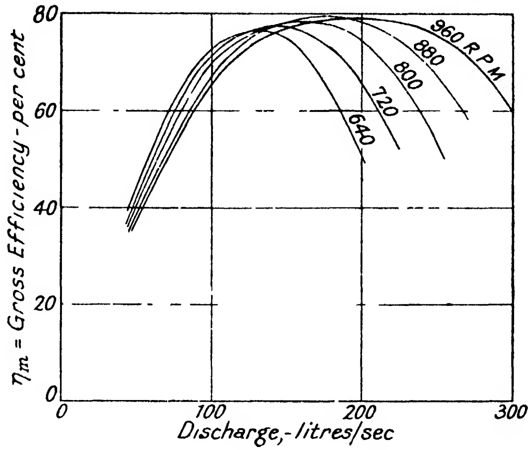


FIG. 150. Discharge efficiency characteristics, corresponding with Fig. 148.

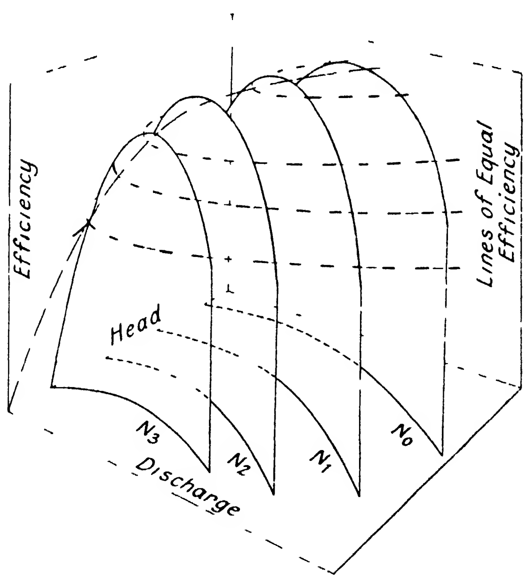


FIG. 151.—Three dimensional plot of complete pump performance.

the curved *surface* enveloping the efficiency curves: it has the form of a curved ridge whose crest is marked by the trace of the successive "equivalent design points". At one end this crest

slowly rises to the summit which represents the maximum efficiency at the design point; at the other end the crest rapidly falls in a kind of arête to zero level or point of zero head.

To represent this three-dimensional surface on a two-dimensional diagram, we use just such a system of contour lines as serves for showing on a map the shape of a topographical surface. Here the contour lines, Fig. 151, will be lines of uniform efficiency instead of uniform altitude. When they are projected vertically downwards on to the head-discharge diagram, they are given the name *iso-efficiency curves*, and the resulting composite diagram has the form shown in Fig. 152. In an ideal pump in which the variable influences described in

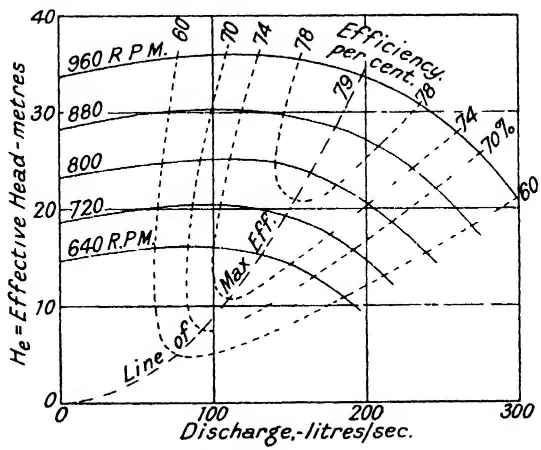


FIG. 152.—Complete characteristics for centrifugal pump, including iso-efficiency curves. (See Figs. 148, 149, 150.)

§ 219 could be disregarded, the iso-efficiency lines would themselves be parabolic, ranged on either side of the parabola that marked out equivalent design conditions, and all meeting at the origin or zero point of the co-ordinates. In fact these parabolas are distorted into the shapes indicated in the figure.

The perspective model, Fig. 151, is needed for illustrative purposes only. The construction actually used for developing iso-efficiency curves is as follows: drawing across Fig. 150 a horizontal line representing an efficiency of (say) 60 per cent., we note the rates of discharge at which this line cuts the efficiency curves, at various speeds, and we transfer the resulting values to the corresponding head-discharge curves,

Fig. 148. A smooth curve drawn through this set of points then yields the 60 per cent. iso-efficiency line. By repeating the process for other values of efficiency, we obtain the complete series, Fig. 152. **(Examples 24, 25)**

223. Performance at Standard Speed. There is still another means of condensing the information provided by Figs. 148 and 149. Instead of plotting *observed* values of head, discharge, and power at a stated speed, we might plot *computed* values showing what the head, etc., would be at some chosen standard speed—say a speed of 1000 r.p.m.—always assuming

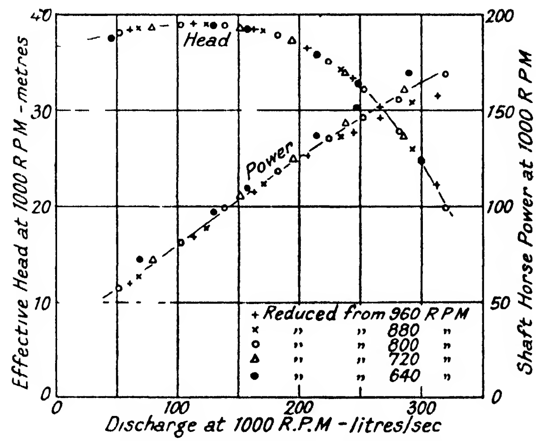


FIG. 153.—Centrifugal pump performance at standard speed of 1000 r.p.m., transposed from Figs. 148, 149

that the appropriate ideal relationships remained applicable, §§ 53, 218. Let the observed values at a speed N be H, Q, and P. Then according to the ideal laws of affinity, the plotted values would be $H \times \left(\frac{1000}{N}\right)^2$, $Q \times \left(\frac{1000}{N}\right)$, and $P \times \left(\frac{1000}{N}\right)^3$. By this method the mean curves in Fig. 153 have been prepared from the data used for Figs. 148 and 149. The manner in which the individual points lie on the mean curves or noticeably drift away from them forms a highly illuminating commentary on what was said in §§ 219, 220. By reason of the accurate alignment of the " head " points we may infer that the affinity law $H \propto N^2$ is very nearly realised in practice. The scatter of the " power " points, on the other hand, is clear evidence that

the true value of the exponent n in the expression $P \propto N^n$ is slightly less than the ideal value 3.

For the daily work of the engineer and designer, the characteristics directly plotted in Fig. 152 are more likely to be useful than the derived curves of Fig. 153.

BEHAVIOUR OF GEOMETRICALLY-SIMILAR PUMPS

224. Effects of Change of Size. In the ideal conditions postulated in § 54, it was found that pumps of the same geometrical shape, but of different sizes, would generate *equal* heads at *equal* efficiencies if :—

$$N \propto 1/D,$$
$$Q \propto D^2.$$

As before, the symbol D is generally representative of rotor diameter, and the stipulation of uniformity of shape means that all the rotors to be compared necessarily have the same specific speed or the same shape number.

For pumps of the same uniform shape but of *any* size, running at *any* speed, the laws to ensure constant maximum (ideal) efficiency were exposed in § 55, viz. :—

$$Q \propto ND^3,$$
$$H \propto N^2D^2,$$
$$\text{whence } P \propto N^3D^5.$$

Before applying these laws to actual pumps under working conditions it will certainly be necessary to examine the relative power losses, as in § 219. But there must also be another enquiry. Is it really possible to attain true geometrical similarity ? In a very carefully built scale model the departures from perfect resemblance may only be trifling, but it is most unlikely that a small commercial pump will be a correct replica of a large pump of the same nominal shape. In comparing the efficiencies of the two machines, the effects of any such lack of similarity cannot be ignored.

225. Relation between Size and Efficiency. Instead of following step by step the line of argument developed in §§ 219, 220, we may profitably choose a more general treatment here (*). In a very broad sense, it seems likely that the gross relative power loss in a rotodynamic pump is linked with a type

of Reynolds number having the form $\left(\dfrac{v_2 D}{\nu}\right)$. This comprehensive dimensionless expression includes terms representative of (1) rotor velocity, (2) rotor diameter, (3) kinematic viscosity of liquid.

Guided by experimental evidence, we may provisionally write :—

$$\text{Gross relative power loss } \Delta_t = k_{13}\left(\frac{v_2 D}{\nu}\right)^y,$$

$$\text{or gross efficiency } \eta_m = 1 - k_{13} \cdot \left(\frac{v_2 D}{\nu}\right)^y, \quad . \quad \textbf{(15-1)}$$

where the exponent y may have an average value of about − 0·2. At least these expressions tell us concisely what is in fact true, that either (i) *reducing* velocities in the pump, or (ii) *reducing* the rotor diameter, or (iii) *increasing* the kinematic viscosity of the liquid, will *reduce* the pump efficiency.

Since rim velocity v_2 is proportional to ND, and since the value $y = -0{\cdot}2$ can be accepted for present purposes, the above expressions can be transposed thus :—

$$\text{Gross efficiency} = \eta_m = 1 - \frac{K}{(ND^2)^{0\cdot 2}} \quad . \quad \textbf{(15-2)}$$

where K is a factor which is *constant* for a given shape number and liquid. The value of this factor K can only be found by actually measuring the pump efficiency ; but when once a figure has been obtained for one pump, the efficiency of any other geometrically similar pump could be forecast with the help of a graph such as Fig. 154. This is a plot of equation (15-2), adapted for a range of pumps in which the speed N is held steady.

226. Application to Model Tests. It is in interpreting the results of model tests, §§ 183, 184, that the above formulæ are likely to be particularly useful. From the data collected while the scale model pump is on the test-bed, how can the performance of the full-scale prototype be predicted ? There would be little difficulty if the formulæ were clear-cut and consistently reliable. But they are not : we may not even know the value of the constant K, although we are fairly sure that it varies from one type of pump to another.

If tests on one size only of model are possible, it follows that

considerable judgment and experience must be brought to bear before much guidance can be extracted from the test results. If models of *two different sizes* were tested, reliable forecasts might then be within sight; having established two points on a curve such as Fig. 154, it would be reasonably safe to extrapolate up to full-scale conditions. Of course, the ideal combination would be a comprehensive series of tests on models, intended to settle the shape of pump that would give the desired characteristics, followed by one final efficiency test on the finished full-scale pump. **(Example 26)**

Some comments upon the most suitable *scale* to adopt were given in § 184. Here it is worth remembering what was implied in § 224—that the larger the model the more nearly can true geometrical similarity be achieved. Experience suggests that the scale should lie between $\frac{1}{6}$ and $\frac{1}{3}$: the diameter of the model rotor should be anything from $\frac{1}{6}$ to $\frac{1}{3}$ that of the full-size rotor.

227. Application to Series of Pumps. In a range of standard pumps built for sale and subsequent use in normal working conditions, it is no longer reasonable to expect that they will exactly fulfil the laws of geometrical similarity. Almost certainly the relative roughness of the passages of rotor and casing will be greater in a small pump of the series than in a large pump. In the small pump the rotor blades may be *relatively* thicker than in the large pump; similarly the clearance between the impeller eye and the casing may be *relatively* greater. All such discrepancies tend to increase the total relative power loss of the small pump *beyond* the point inevitably fixed by the change in size of the pump, § 225. This implies that the curve between rotor diameter and efficiency will fall rather faster than is shown in Fig. 154. Alternatively, we might say that the equation between efficiency and size given in § 225 should be modified thus :—

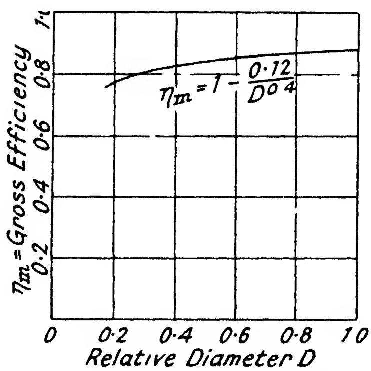


FIG. 154.—Typical relation between diameter and efficiency.

$$\eta_m = 1 - \frac{K}{N^{0.2}D^{0.45}}. \qquad . \qquad . \quad (15\text{-}3)$$

It is such trends as these that a maker must rely upon when the range of pumps of a given general shape number is to be extended to larger sizes. That these tendencies might be expected was shown in Fig. 53; there the size of the pump was expressed in terms of its power output, § 91, but since the output of a rotor of a given shape ultimately depends upon its speed and diameter, it follows in a general way that Figs. 53 and 154 are in harmony.

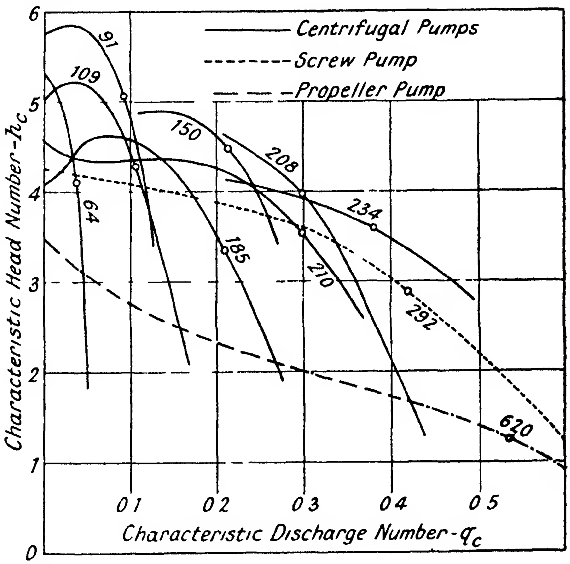


FIG. 155 Non-dimensional plotting of head discharge characteristics for various types of pumps The numbers give the shape number at the respective design points shown by circles

228. Universal Non-dimensional Characteristic Curves. Just as a single curve in Fig. 153 served to show the head-discharge relationship for a *given* pump running at any speed, so we may now plot a curve applicable to *any size* of rotor (of a given shape) running at any speed. It will be still further illuminating if we collect on a single sheet a number of these curves, Fig. 155, in such a way as to show how rotor shape affects rotor performance. Each curve is based on test figures taken from an actual pump. As representative of the head we

may choose the head developed by a rotor of unit diameter running at unit speed, $\dfrac{H}{\left(\dfrac{N}{60}\right)^2 D^2}$; or still better, the *characteristic head number*, $h_c = \dfrac{gH}{\left(\dfrac{N}{60}\right)^2 D^2}$, § 56. The corresponding term representing discharge is the *characteristic discharge number*, $q_c = \dfrac{Q}{\left(\dfrac{N}{60}\right) D^3}$. Here are graphs that, within their own limits, are truly universal; not only are they independent of size or speed, but because they are expressed in non-dimensional terms they are independent also of the units employed, so long as these are consistently chosen, § 56. Naturally the curves cannot take into account the slight departures from the basic affinity laws examined in §§ 220, 225, and 231.

The chief value of Fig. 155 is that it reinforces so graphically the information presented in Fig. 35. The earlier diagram showed how rotor shape influenced pump performance *at the design point only*; the present diagram extends the range of information to cover any region of performance. **(Example 27)**

Influence of the Working Medium on Pump Behaviour

229. Types of Working Medium. The term "working medium" is here used to describe in the broadest sense the substance actually passing through the pump passages at any moment. The word "liquid" or even "fluid" might have too restricted a significance. It is true that most pumps are supplied with liquid only—usually water. But sometimes the liquid has a considerable load of solids, § 146; or it may be mixed with air or with vapour, § 249. For special purposes the pump may be run with nothing but pure air flowing through its passages, §§ 236, 237. Just as paragraphs in Chapter X described modifications in design to meet such unusual conditions, so the following paragraphs indicate how the performance may be altered. As a standard of performance by which the modified performance may be judged, it will often be convenient

to accept the behaviour of the pump when handling (nominally) pure water at atmospheric temperatures. With the pump running *at a selected speed and rate of discharge*, what will be the effect on (*a*) the head generated, (*b*) the pressure-difference generated, (*c*) the power input, and (*d*) the efficiency, if the clean water is replaced by the specified working medium ?

230. Effect of Change of Density. Let us first assume that the viscosity of the new liquid is substantially the same as the viscosity of water or at least that the difference between the two will not sensibly alter the relative friction losses in the pump passages, § 231. At once we can answer question (*d*) : we can say that the pump efficiency will *remain virtually unchanged.* Moreover, since all velocities remain as they were, whether of the rotor or of the liquid, it follows that the *head* also remains constant. But since pressure = head × density, § 165, we see that the pressure difference p_e will change in proportion to the change in density.

If subscripts $_w$ relate to water,

,, $_l$ relate to working liquid, then

$$\frac{p_{el}}{p_{ew}} = \frac{w_l}{w_w} = \frac{\text{density of liquid}}{\text{density of water}} = \text{S.G. of liquid.}$$

In regard to *power input*, P_s, the relationships are :—

$$\frac{P_{sl}}{P_{sw}} = \frac{p_{el}Q}{p_{ew}Q} = \frac{w_l}{w_w}.$$

Thus, both the input power and the output power for a given *volumetric* discharge vary in proportion to the density.

The general effect of these changes is depicted in Fig. 156, where comparative characteristics for a two-stage pump are plotted

231. Effect of Changes of Viscosity. Considering now a series of liquids of progressively increasing viscosity, ranging from water or spirit up to thick, heavy oils, we may expect to reach a point at which the pump is no longer insensitive to viscosity changes. In taking into account the subsequent effects, which will doubtless become more and more marked as the liquid grows thicker, we cannot ignore the influence of density changes ; the two effects must both be recognised, and in fact they nearly always operate simultaneously. Liquids that are much more viscous than water almost invariably have

a density greater than or less than that of water. In this paragraph and the companion ones, the influence of viscosity is to be studied separately. Taking a broad, practical view, the onset of the resulting deterioration of performance can be divided into three phases :—

(i) Small increases of viscosity, in relation to the viscosity of water which is taken as standard, produce only a negligible effect on the pump performance, § 230.

(ii) Larger increases have little effect upon the *head* generated, but the pump *efficiency* noticeably declines.

(iii) With still more viscous liquids, both the head and the efficiency are seriously reduced in relation to their values when water flows through the pump.

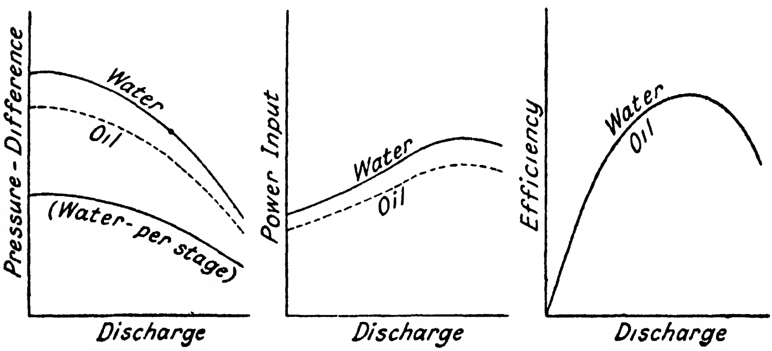


FIG. 156 — Comparison between pump working with water and working with oil (Discharge plotted *volumetrically*)

No precise limits can be assigned to these successive stages. In the very roughest way, it may be said that if the pump impeller is about 1 ft in diameter, then phase (i) may end when the liquid viscosity has risen to about 0·1 poises, and phase (ii) when the viscosity reaches 0·3 poises. That is to say, phase (ii) relates to liquids that are between ten and thirty times as viscous as cold water (*)

In trying to explain these observed effects, why should we not apply the general equation for pump efficiency (15-1), developed in § 225 ; for the influence of viscosity on efficiency was there clearly foreshadowed ? The reason is that the proportional changes now in question are far more extensive than they have hitherto been. Variations in speed or diameter have

only involved a ratio between minimum and maximum values of perhaps 1 to 6; but we may have to deal with a range of viscosity variations of the order of 1 to 100 or 1 to 1000. Indeed, a single diagram, e.g., Fig. 157, may embrace information about the behaviour of liquids whose kinematic viscosity varies from 0·01 stokes to over 40 stokes. It is hardly likely that such

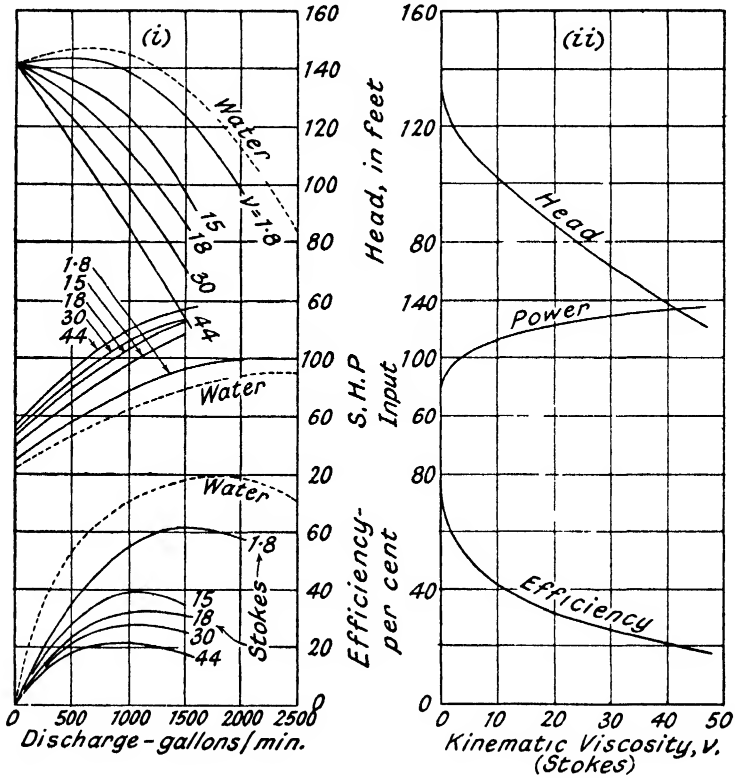


FIG. 157.—Graphs showing how changes of viscosity of liquid affect pump performance

a diversity of liquids will follow a common law while traversing the pump passages; sometimes they obey the turbulent regime of flow and sometimes the laminar regime. Whatever correlation there may be, then, between Reynolds number and pump performance, it will almost certainly be too complex to be of much use to the engineer. At present he must rely upon a cruder analysis of experimental data. (See also § 190 [5].)

232. Changes in Pump Characteristics. To find out how a given pump responds to viscosity variations, the most effective method is to run the pump first on water, and to take from it a complete set of constant-speed characteristics. Then another liquid—say a thin oil—is substituted for the water, and another set of curves plotted. Thicker and thicker oils are in turn tried, always with the pump speed held steady, until the performance has fallen far below what would be tolerated in practice. When all the resulting graphs are plotted on a single sheet, the picture has some such form as Fig. 157 (i); the centrifugal pump here in question has 8-in. branches and a 15-in. diameter impeller.

As we should expect, the curves show that for a given speed and rate of discharge, the head and the efficiency decline as the viscosity increases, while the S.H.P. input rises. It is true that the power curves are not in the strictest sense comparable; they include the effects both of viscosity *and* of density, § 231. But their trend is clear enough(*).

The *rate* of change of the variables can more clearly be perceived by plotting them directly against kinematic viscosity, as in Fig. 157 (ii). Suitable values are here directly transferred from diagram (i), relating to a selected rate of flow of 1500 gall./min. The curves show what degree of importance should be attached to the notion of "phases" used in § 231. Although the range of viscosities is so great that phases (i) and (ii) cannot properly be discerned, yet there is nothing in Fig. 157 (ii) to hint that there are any discontinuities in the correlation between viscosity and performance. Changes in performance probably extend right down to the lowest limits of viscosity; the conception of phases is purely arbitrary, serving merely to indicate when certain variations become perceptible.

233. Combined Effect of Viscosity and Diameter. From the reasoning of § 231, we may infer that the more viscous the liquid, the more sensitive will the performance become to variations in impeller diameter; that is to say, the type of variation studied in § 225 will become more pronounced. For instance, if the experiments summarised in Fig. 157 were to be repeated with a smaller pump, but using the same kinds of oil, it seems likely that the characteristics would diverge still more widely from the standard water characteristics.

(Example 28)

Performance figures supporting this inference are plotted in Fig. 158. They have been abstracted from test results on a range of pumps having impellers of various sizes, and of shape numbers between about 60 and 120. The oils used ranged in kinematic viscosity from 0·25 stokes to 4 stokes. Estimated average values of head and efficiency, at the rate of discharge corresponding to the design point for water, are plotted against impeller diameter in the manner proposed in § 229, viz., as a percentage of what the equivalent values would be with water. Because the pumps concerned are not geometrically similar, but have the shape usually associated in practice with the respective impeller diameter, the graphs are indicative of general trends only rather than of precise laws(*). Nevertheless

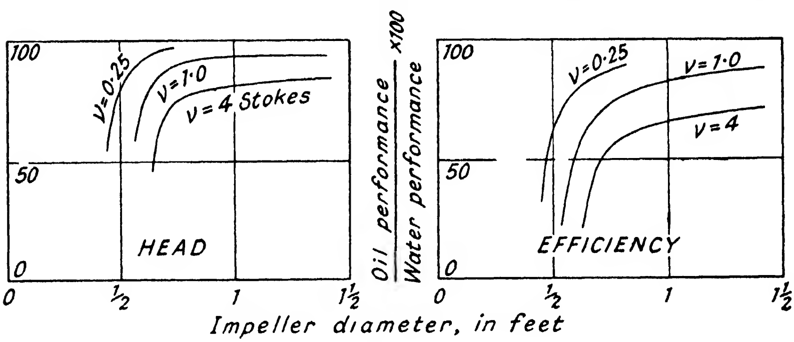


FIG. 158.—Comparative pump performance under conditions of varying shape number, varying impeller diameter, and varying liquid viscosity.

the lesson they teach is clear enough. To achieve a tolerable efficiency when pumping liquids of high viscosity, *only a relatively large pump* will serve. For handling small quantities of such liquids, centrifugal pumps are *radically unsuitable.*

234. Other Considerations. (i) *Speed.* Undoubtedly the efficiency of pumps handling viscous liquids is susceptible to speed changes, but the effect is now far too complex to fall within the relatively simple treatment used in § 220.

(ii) "*Universal*" *Characteristics.* As the liquid viscosity rises, so must we lose hope of plotting any such "universal" characteristics as were presented in Fig. 155. In order to predict with complete accuracy the performance of any given pump, it would really be necessary to plot for each liquid and for each size of pump a separate family of constant-speed

characteristics. But a range of graphs of some such form as Figs. 157, 158 usually gives guidance for many industrial purposes.

(iii) *Temperature.* Although temperature changes have not so far been specifically mentioned, yet by their effect on liquid density and viscosity they manifestly exercise a highly important influence on pump behaviour. The density of *water* falls by more than 4 per cent. as its temperature changes from freezing-point to boiling-point; while if the water attains the temperature often found in steam boilers, the reduction in density may be 15 per cent. or more. These changes, of course, affect only the power input to a pump and the pressure difference generated, § 230. The density of *oils* and liquid organic compounds generally is much more susceptible to temperature changes; but it is the corresponding changes in viscosity that have the predominating influence on pump performance. A temperature rise of a few degrees only may sometimes halve the liquid viscosity.

235. Effect of Suspended Solids. When water carries with it a load of suspended solids, § 146, the mixture behaves as though it were a new liquid of greater density and greater viscosity than pure water. Such mixtures will affect the pump behaviour much as viscous liquids do, §§ 231 to 233; but because of the diverse nature of the constituents of the mixtures it is not practicable to frame even such general rules as served for those pure liquids. Thus, water charged with say 5 per cent. of wood-pulp or paper stock is quite difficult to pump economically; with such a mixture the pump efficiency may be only 50 per cent. of the corresponding value with pure water, and to achieve even this moderate result the discharge must be reduced much below the "design" water discharge.

It must be remembered, too, that the original design of the pump passages, § 149, may necessarily have been modified in such a way that the basic efficiency is already impaired. An extreme example is provided by the pumps used in suction dredgers. The impeller may have only four radial blades, adapted to a mud-and-water mixture whose specific gravity may rise to 1·6.

236. Pumps working with Air. If the correct relation between speed and discharge is maintained, a pump running on

atmospheric air may still remain a satisfactory machine judged by rotodynamic standards. It still generates its stipulated head: but now it is "head" of air and not of water. As § 230 shows, the ratio between the corresponding pressures has a value of about $\frac{1}{800}$. Many pumps when running at their normal speed would thus generate quite an insignificant pressure difference—as fans or blowers they would be judged ineffective —but sometimes on the test-bed useful information can be gathered in favourable conditions by measuring this pressure difference, § 183. Such tests reveal fairly accurately the shape of the head-discharge characteristic, but are of little value in predicting the power that would be required with the working liquid. The reason is clear: on test, the output power is only about $\frac{1}{800}$ of what it would be under working conditions, while on the other hand the mechanical power loss due to bearing and stuffing-box friction remains virtually unchanged. This means that during the air test the *relative* mechanical power loss, $\triangle_b$, would be disproportionately high. In any event it might be difficult to measure with sufficient precision the comparatively small power input involved.

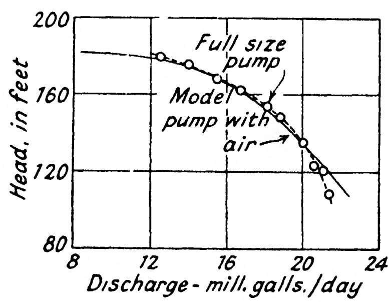


FIG. 159.- Comparative performance with air and with water.

Models specially built for testing with air may be reproduced either full-scale or to a reduced scale. As the casings may now be of wood or plaster or similar easily-worked material, modifications as dictated by successive tests are even simpler to carry out than when water-test models were used, § 226. The very close accord between the two head-discharge characteristics plotted in Fig. 159 shows how helpful this technique can be (*). One of the curves relates to a full-scale pump working on water; the other to a model one-third full-size, working on air.

237. Model Pumps working with Compressed Air. Comparing compressed air with atmospheric air as a testing medium for scale-model pumps, we at once discern two very clear

advantages : (1) Because of the greater density of the compressed air the pressure difference generated by the model pump is sufficiently high to be measurable with considerable precision, (2) similarly, reliable observations of power input can now be taken, especially as the special construction of the model will greatly reduce the problem of stuffing-box friction. But there is still another advantage. As the air pressure rises, the kinematic viscosity of the working fluid *falls* ; and if the speed of the model is kept steady, the result will be that the Reynolds number $\left(\frac{v_2 D}{\nu}\right)$, § 225, will *rise*. According to earlier reasoning, the efficiency of the model might be expected to rise also.

(Example 29)

In order to profit by this relationship, the test procedure is arranged thus : The model is mounted in a closed circuit as described in § 183, taking care that a cooler is interposed to abstract from the air the heat corresponding to the energy continually fed into it. Throughout the test, the model is run at a constant speed, as fast as is safe or convenient. Successive tests are made with progressively rising air pressures, until a maximum of 5 to 10 atmospheres is reached. When efficiencies are plotted against absolute pressure, the curve has the form indicated in Fig. 160 ; the expected upward trend is fully realised (*). The regular shape of the curve, moreover, justifies its extension, as shown by the broken line, to the point corresponding with the Reynolds number of the full scale pump when working normally with water. Reading off against the efficiency scale the equivalent value of the model efficiency at that point, one could finally say that that value also represented the efficiency of the prototype pump under its own design conditions.

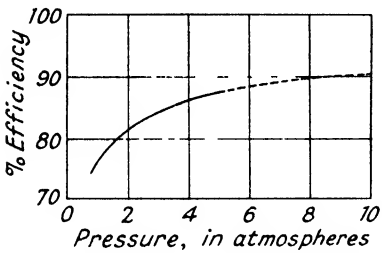


FIG. 160. Influence of air pressure on efficiency.

As a first approximation to the truth, such an inference might be allowed to pass. But it would not satisfy a pump maker facing the responsibility of predicting the performance

of a large and costly pumping installation. Nevertheless the information yielded by the model tests, *supported by all other available information,* would be extremely useful. Here it remains to be pointed out that the curve in Fig. 160 is basically similar in shape to the efficiency curves of Figs. 147 (ii) and 154, as it should be if the reasoning of § 225 is acceptable.

CHAPTER XVI

INSTALLED CONDITIONS

238. Test Conditions and Installed Conditions. The title of this chapter at first looks a trifle discouraging ; it suggests that after the pump has been installed on site and has begun its regular work, it may behave differently from what it did on the test-bed. There may indeed be such a difference, but more often than not the observed variations concern the installation *as a whole* rather than the pump itself. The variations to be studied can be classified thus :—

(i) *Effect of Pipe Circuit or Lay-out.* Figs. 107 and 108 have already shown how a slight change in the external circuit can influence the flow through the pump. If, in the simple systems there illustrated, the static head had remained unchanged, nevertheless the interpolation of the lengths of pipe would certainly have reduced the pump discharge and increased the head on the pump, although the pump speed had not altered. As we have now learned to say, the pump performance would have moved to another point on its characteristic. It is our present purpose to examine all such possibilities of change of regime, whether caused by pipes, valves, or other pumps.

(ii) *Effect of Suction Head.* Hitherto the effect of the suction head on the pump has been completely disregarded ; apparently the pump performance, as revealed by the pump characteristics, depends only upon the total or effective head H_e, § 162, and

not at all upon the height of the pump above the suction well. Yet there must ultimately come a stage at which the static suction lift h_s, or the manometric suction head H_{ms}, begins to control the pump behaviour, because we are quite certain that if the pump were set more than a limiting distance above the suction-well water surface, the atmospheric pressure would never feed the liquid up to the pump at all. It is essential to know what those limits are, and to learn what happens if they are approached or transgressed.

(iii) *Effect of Age or Length of Service.* Mechanical wear, or other kinds of erosion or corrosion after prolonged service, may perceptibly alter the shape or condition of the working parts of the pump. Deterioration of performance will then certainly be expected.

A sharp distinction is thus to be noted between category (i), and categories (ii) and (iii). The external circuit, (i), does not interfere with the internal working of the pump; the relation between pump discharge, and effective head *as measured across the pump branches*, follows exactly the law that was revealed on the test-bed. But in conditions (ii) and (iii) the actual characteristics may be distorted, and thus the test figures do not necessarily give reliable guidance on what the pump will do. These two conditions, moreover, may interact one upon the other. If the suction lift is excessive, the decline in pump efficiency implied in (iii), above, may progress much more rapidly than it otherwise would.

Influence of External Circuit

239. Methods of Controlling Head or Discharge. In practice we frequently have to depend upon intentional changes in the external circuit for bringing about desired variations in the head or discharge of the pump (*). Some possibilities are :—

(i) With the pump running at *constant* speed, we can regulate the hydraulic resistance in the external circuit by adjusting a throttle-valve, by changing liquid levels, etc.

(ii) In *variable-speed* pumps, there may be another relationship between performance and external resistance.

(iii) The *number* of pumps effectively in circuit can be altered, e.g., two or more pumps can be disposed in series or in parallel.

(iv) Combinations of these systems can be contrived.

In choosing between various systems of regulation or between various combinations, the guiding rule should be that the pump efficiency must consistently be maintained as high as possible. Manifestly the best method would be to keep the pump running always on the line of maximum efficiency, § 220, but it is very rare that this programme would be flexible enough to meet normal requirements. Nevertheless we can try to keep near the maximum-efficiency " ridge ", Fig. 151.

In making a detailed analysis of methods of regulation, we naturally begin with those applicable to constant-speed pumps, because electric drive so often implies a fixed speed or a fixed range of speed. Graphical plotting of the variables is helpful, especially if the same vertical scale is used for representing both the hydraulic system itself, and the pump head-discharge characteristic. This has been done in the following diagrams.

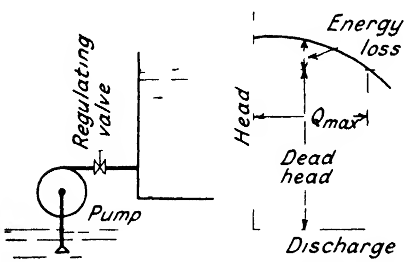


FIG. 161.—Discharge regulation by throttling

240. Constant Speed, Constant External Head, Variable Discharge. (i) *Dead Head Only.* The pump discharges through a short pipe, of assumed negligible resistance, into a constant-level reservoir, Fig. 161 ; in order to meet service fluctuations the flow is controlled by a throttle-valve. Maximum flow, $Q_{max.}$ occurs with fully-opened valve, when the pump effective head is virtually identical with the dead head. As the valve is progressively closed, the flow diminishes, the effective head rises in accordance with the shape of the pump characteristic, and *more and more energy is wasted in the valve.* That is an important point ; if the valve is not fully open, and if the pump is not working at its design point, there is a double loss of energy—in the pump itself and in the valve. The loss

can be reduced by judiciously controlling the shape of the head-discharge characteristic, § 214 ; the flatter the curve, the less the waste of energy in the valve. Before finally accepting a likely form of curve, it is essential to make sure that the closed-throttle head is greater than the dead head, otherwise the pump would never be able to initiate flow at all.

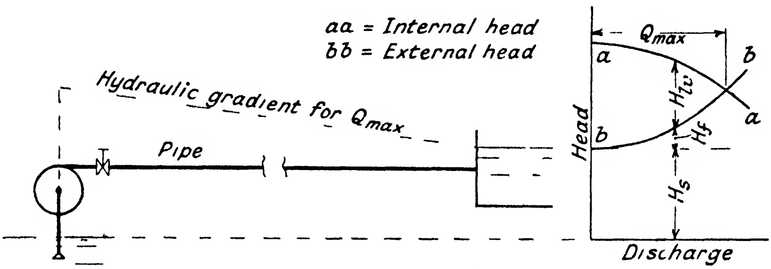


FIG. 162.—Constant speed pump discharging through long pipe.

(ii) *Dead Head and Friction Head.* When the pump forces liquid into a constant-level reservoir through a long pipe, Fig. 162, the external resistance may now comprise three items, (1) the dead or static head, (2) the energy loss in the valve, (3) the frictional loss in the pipe. Denoting these respectively by H_s, H_{lv}, and H_f, we can equate the "internal" head H_e generated by the pump to the "external" resistance to be overcome, in this approximate manner :—

$$H_e = H_s + H_{lv} + H_f.$$

Now the value of the frictional term H_f depends very nearly upon the square of the discharge Q, and it can thus be represented by the parabolic curve *bb* in Fig. 162. Evidently, then, the maximum discharge through the system with fully-open valve will correspond with the intersection between this curve and the pump head-discharge characteristic, *aa*. For smaller flows, the loss in the regulating-valve builds up rapidly : it is bound to be relatively more serious than the equivalent loss in the simple dead-head circuit, Fig. 161.

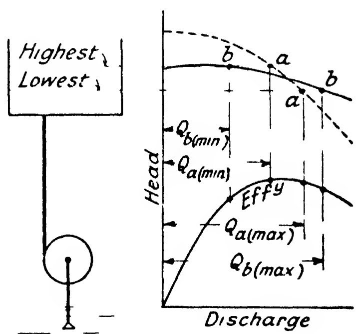


FIG 163.—Effect of tank water level on pump performance.

Before the least wasteful type of pump characteristic can finally be specified for either of the above layouts, it is essential to have some notion of what may be called the load-factor of the system. If this factor is high: if the demand for liquid is always relatively great: then the design point of the characteristic should be fairly close to the maximum-discharge point. But if full flow is only rarely wanted, then it may be advantageous to shift the design point towards the mean flow point of the system.

241. Other Constant-speed Systems. (i) *Constant Speed; Head and Discharge Controlled by Liquid Level.* If a constant-speed pump is used for lifting liquid from one tank to another, and it is no longer necessary to regulate the rate of flow, then waste of energy in a valve can be eliminated (Fig. 163). To ensure minimum overall power consumption, the pump should preferably be arranged to work near its design point when the tanks are half full. Here is an instance where a steeply-falling characteristic, *aa*, is advantageous, for it implies a fairly uniform rate of flow throughout the whole pumping cycle. To use a flatter characteristic, *bb*, would mean not only a large variation in rate of discharge—which in itself might be unobjectionable—but also a marked decline in efficiency at the two extreme points of the cycle. The comparative diagrams in Fig. 163 make this point clear. They show how the rate of flow steadily falls away as the upper tank fills. At the lowest tank level, viz., at the beginning of the pumping period, the discharge is $Q_{max.}$; at the end of the period, when top water level has been reached, the discharge is only Q_{min}. **(Example 31)**

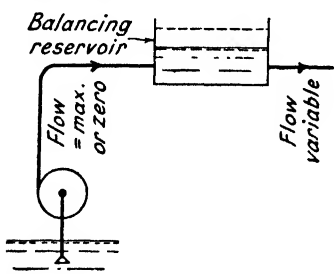


FIG. 164.—Use of balancing reservoir for improving performance.

(ii) *Constant Speed, Constant Head, Intermittent Pump Operation.* Fig. 164 shows a system which enables constant-speed, constant-head pumps to work near their design point and yet to give a variable discharge along a pipe. The pump either runs under conditions of (virtually) full efficiency, or it is stopped; a balancing tank or reservoir maintains flow along the pipe during the idle periods. When the demand for water

is high, these idle periods are brief and infrequent ; when there is only a small demand, it is the pumping periods that are curtailed. It is true that the working head on the pump is only nominally constant, but the fluctuations can be kept very small by suitably proportioning the balancing reservoir. In no other type of large pumping installation can so low an overall relative energy consumption be guaranteed.

(iii) *Constant Speed, Constant Head, Variable-discharge Pump.* The propeller pump alone has the advantage of being able to deliver a variable discharge in these circumstances without suffering serious loss of overall efficiency. Its construction was described in § 104, and its form of characteristic was explained in § 216 (*b*).

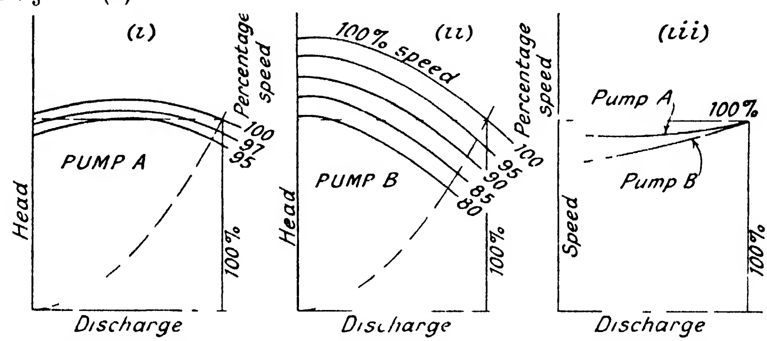


FIG. 165.—Influence of pump characteristic on discharge regulation, with varying speed, against constant dead head.

242. Variable Speed, Variable Discharge. (i) *Dead Head only.* The conditions described in § 240 and illustrated in Fig. 161 can now be met without the use of a regulating valve and without the consequent energy loss therein. But at low rates of discharge, which are here attained by lowering the pump speed, § 221, the pump is still working under unfavourable conditions, Fig. 165. Diagrams (i) and (ii) represent the head-discharge performances of two pumps, *A* and *B*, while diagram (iii) shows how speed variations influence the rate of discharge. These graphs suggest that whereas formerly a flat characteristic was preferable, pump *A*, we ought here to choose a sloping one, pump *B*. Not only will the flat type almost certainly be unstable, § 215, and not only will the instability be unchecked by the steadying influence of the throttle-valve, but

in any event the whole system will respond too sensitively to speed variations. As is evident from the graphs, it may happen that the discharge would be materially affected even by the slight rise in speed consequent upon the normal heating up of a direct-current motor ; a 5 per cent. rise in speed may produce a 30 per cent. increase in discharge. **(Example 30)**

(ii) *Dead Head and Friction Head.* Whereas this combination showed the constant-speed pump at its worst, Fig. 162, it enables the variable-speed pump to behave at its best. In the limiting case of zero dead head, then the ideal conditions postulated in § 239 can very nearly be attained : the parabolic internal curve of maximum efficiency virtually coincides with the parabolic external curve, Fig. 166 (i). Even if the dead head is quite substantial, Fig. 166 (ii), there remains quite a wide range of discharge within which the departure of one

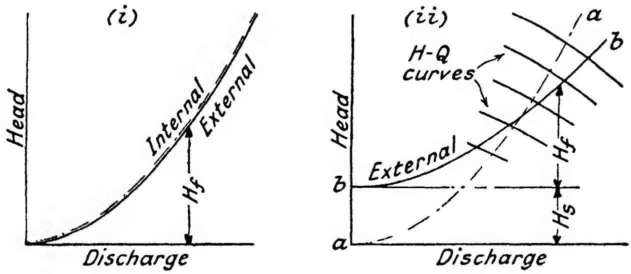


FIG. 166.—Relation between internal head and external head for variable-speed pump.

curve, *aa*, from the other, *bb*, is insufficient to cause any noteworthy decline in efficiency. As in all these diagrams the discharge prevailing at any particular speed is read off from the intersection between the external head curve and the appropriate pump head-discharge characteristic.

(iii) *Variable Speed, Constant Torque.* At least within a limited range of conditions, the pump is subjected to constant torque—or it can be assumed to be—when it is driven by an internal-combustion engine running at full *engine* throttle. If the head is regulated by throttling of the *pump* circuit, then the head, speed, and discharge will all vary, and their relationship one with the other can be expressed by a new type of characteristic, Fig. 167. The heavy pressures generated at

low rates of discharge are very noticeable, and in practice they may be objectionable.

Fig. 167 relates to the pump whose performance has already been presented in Figs. 148 and 149; it is now assumed to be subjected to a constant torque of 4·6 kilogram-metres. Diagram 167 (i) is directly reproduced from Fig. 149, while the broken characteristics in diagram (iii) are transferred from Fig. 148. Since

$$\text{S.H.P. input} = P_s = \frac{\text{Torque} \times \omega}{k_p} = \frac{\text{Torque} \times \frac{2\pi N}{60}}{k_p},$$

it is easy to use diagram (i) to construct diagram (ii), which consists of a set of torque-discharge curves. From the points at which these curves cut the horizontal line representing the stated torque = 4·6, verticals are dropped to the respective head-discharge curves in Fig. 167 (iii). The desired *varying speed* characteristic *aa* can then be drawn. The speed-discharge curve *bb* can easily be added if required.

The true torque-speed relationship for the engine driving the pump will in fact be slightly curved instead of exactly horizontal, as diagram (ii). But in any event there is no difficulty in correctly setting off in this diagram whatever information is given.

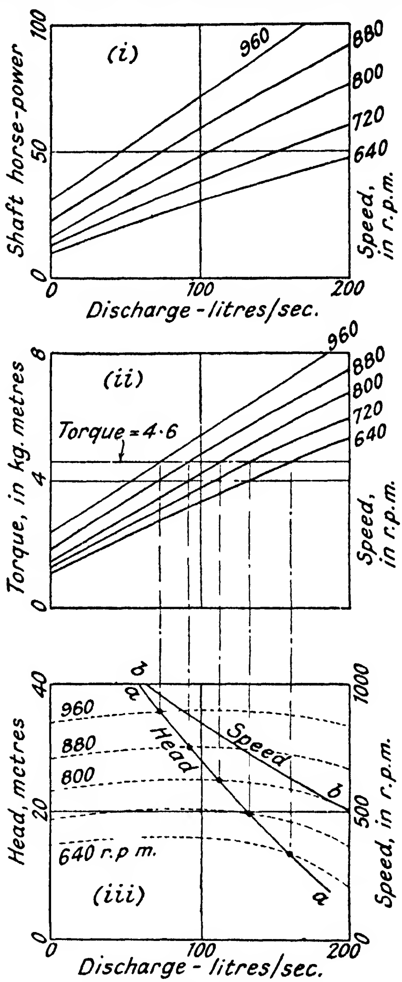


FIG. 167.—Construction for plotting head-discharge, and speed-discharge characteristics, under conditions of constant torque.

243. Use of Multiple Pumps.

(i) *In Series.* When liquid flows in succession through a number of pumps, the basic fact

influencing the flow is this: that at a given moment the discharge *must be the same* in each of the pumps. The numerical value of the discharge can be found by equating the *sum* of the

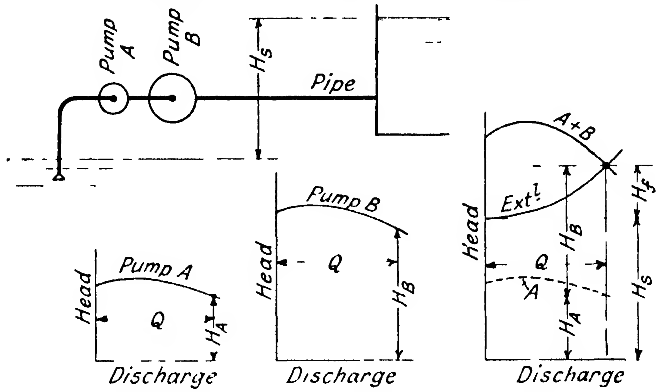


FIG. 168.—Constant-speed pumps working in series.

internal heads to the *sum* of the external resistances. In Fig. 168, two constant-speed pumps, *A* and *B*, are shown, together with their individual head-discharge characteristics and the external circuit characteristic. By the use of a combined head-discharge curve representing the gross head generated, we can read off the desired discharge as in Fig. 162. The individual heads, H_A, H_B, developed by the pumps can in turn be scaled off, and finally the power input to each pump can be computed.

(ii) *In Parallel.* Here the rule is that the pumps must generate *equal* heads, while the discharges must be *added*,

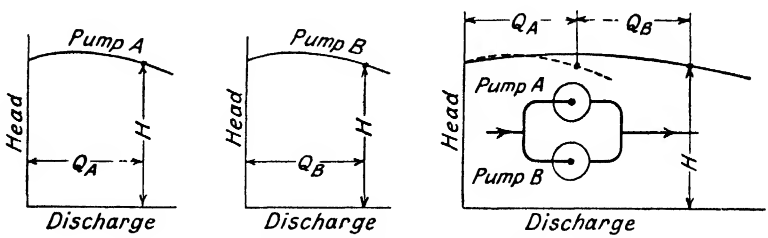


FIG. 169.—Constant-speed pumps working in parallel.

Fig. 169. Parallel operation offers a very effective means of regulating the total discharge: for small total flows, only one pump will be running, and as the demand increases, additional units are started up. By regulating in this way an installation

of three constant-speed pumps, there will be obtained the combined head and efficiency curves shown by full lines in Fig. 170 (✱). Variable-speed pumps give still better results. The broken lines, on the contrary, show how the overall performance deteriorates if pumps are left running needlessly.

(Example 32)

(iii) *Series-parallel.* By arranging a pair of identical constant-speed pumps either (*a*) in series, (*b*) in parallel, a very wide range both of head and discharge is attainable, Fig. 171. Piping systems adapted to these various arrangements are illustrated in § 319.

244. Comparisons between Systems of Discharge Regulation. In order to assess the relative merits of the

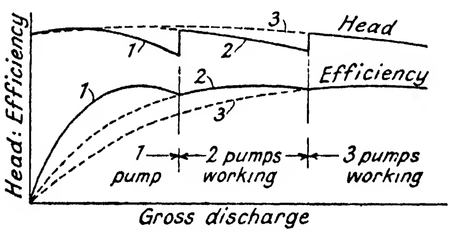


FIG. 170.—Performance of pumps working singly or in parallel. The numbers 1, 2, 3 refer to the number of units in operation.

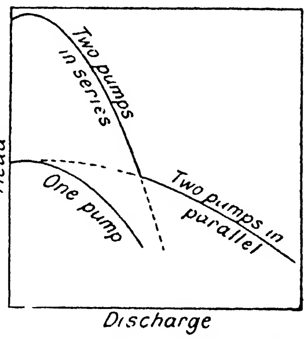


FIG. 171.—Series-parallel operation.

methods of pump control that have been examined, we may conveniently use the notion of *static efficiency* developed in § 165. If we apply it to the dead-head circuit shown in Fig. 161 (pump delivering against constant dead head only), then the resulting graphs, Fig. 172, do give a true picture of the overall effectiveness with which the power fed into the pump shaft is utilised. To put all the pumps on a fair comparative basis, values of static efficiency

$$\eta_{stat} = \frac{\dfrac{WH_e}{k_p}}{\text{S.H.P.}}$$

are expressed in terms of the maximum pump gross efficiency η_m.

Velocity heads are ignored, as they have been throughout this chapter.

As might have been expected from § 240 (i), curve (*a*) is the least satisfactory. The only combination that might give still better results than curve (*d*) would be : to use multiple constant-speed pumps, together with the balancing reservoir shown in Fig. 164.

In all the systems under comparison, it has been assumed that the pumps are working against the hydraulic pressure of liquids in open tanks or reservoirs. Of course the pumps would behave exactly the same, in comparable circumstances, if the

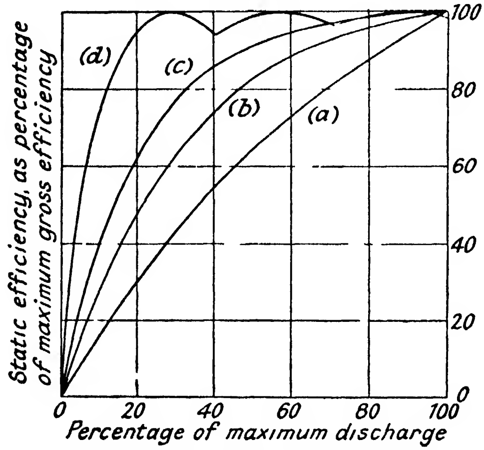


FIG. 172.—Comparison between various methods of discharge-regulation against constant dead head —

(*a*) One constant speed centrifugal pump, throttled.
(*b*) One variable speed centrifugal pump
(*c*) One variable pitch constant speed propeller pump
(*d*) Three variable speed centrifugal pumps in parallel.

pressure were to be maintained by air or steam pressure in closed containers. An example has already been furnished in § 163, Fig. 110.

EFFECTS OF EXCESSIVE SUCTION HEAD

245. Observed Effects. Each of the four identical pumps shown diagrammatically in Fig. 173 is set at a different height h_s above the suction water surface, but otherwise there is no difference between them ; they run at the same uniform speed

and therefore, for anything we have yet learned to the contrary, they should yield identical characteristic curves. As we are here speaking in general terms, velocity heads and pipe friction are ignored. On test it may indeed be found that pumps A and B do behave quite normally. As the throttle-valve is progressively opened, the discharge increases and the manometric head responds as usual, § 212. The characteristic (*AB*) serves equally well for both pumps.

When it is the turn of pump C to be tested, it looks at first as though here also there will be nothing out of the ordinary to report. Beginning with the closed-throttle condition, the successive plotted points fall accurately on the earlier curve

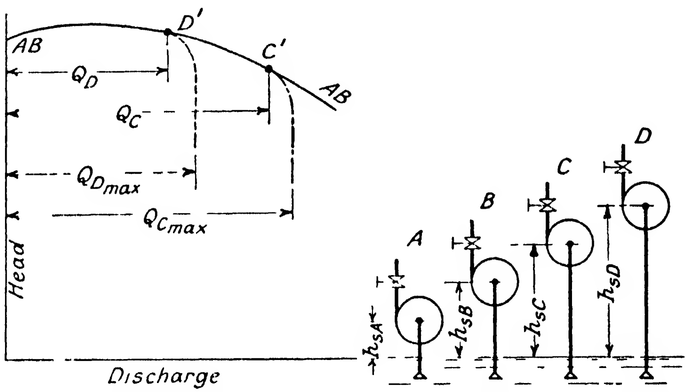


FIG. 173.—Abnormal pump performance under excessive suction lift.

(*AB*). Then, at a certain stage denoted by *C'*, the new points begin to fall *below* the existing curve. As the throttle-valve is still further opened, the new curve grows steeper and steeper, until finally it trends vertically downwards. The maximum attainable discharge has dropped to $Q_{C_{max.}}$, Fig. 173. As we might by this time almost expect, the characteristic relating to the uppermost pump takes its downward plunge still sooner; nothing we can do will induce pump D to give a greater discharge than $Q_{D_{max.}}$. If there were still any doubt that something queer was going on inside the pump, we should only have to listen to it. While the pump was working on the abnormal vertical part of its characteristic, we should probably hear a hissing or roaring sound, or we should notice excessive vibration. Moreover, the test instruments

would show that meantime *the pump gross efficiency had deteriorated* (✱).

246. Critical Suction Head. The abnormalities just described can be defined by taking note of the points such as C', D' in Fig. 173 at which they first become noticeable; these are the points where the distorted characteristics first diverge from the normal characteristic. At any one such point there is evidently a relationship connecting the total head H, the discharge Q, and the static suction head h_{sc}. We might say that this suction head h_{sc} is the *critical suction head* corresponding to a given head H or discharge Q; it is the maximum

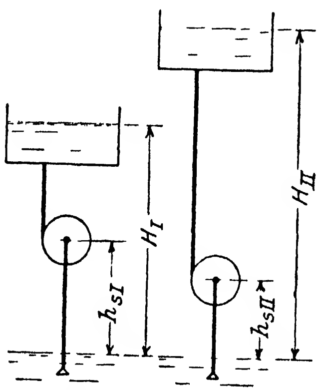


FIG. 174.—Effect of total head on suction head.

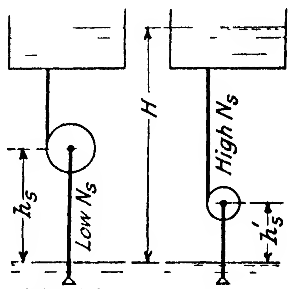


FIG. 175.—Effect of specific speed on suction head.

suction head that could be maintained without exposing the pump to the consequences of abnormal behaviour.

Using this criterion of performance, we can now find out what the original identical pumps in Fig. 173 will do in other circumstances. In Fig. 174 they are shown working against different total heads, H_I and H_{II}, running at different speeds so adjusted that each pump runs at its maximum efficiency point, § 218. The pumps are set at the limiting heights h_{sI} and h_{sII} above the suction well: that is, the critical suction head h_{sc} prevails. The pumps are indeed working normally, but only just—if they were raised only a little higher, the first signs of abnormality would appear. At once we are struck by a most significant fact: the *greater the total head on the pump*, the *less becomes the permissible suction lift.*

Next we may profitably compare two pumps of *different* specific speed working against the *same* total head H, Fig. 175. As before, each pump runs at its own design point and draws against its critical suction head. The diagrams reveal another striking difference : the *higher* the specific speed, the *lower* the suction lift : the height h_s' is *less* than the height h_s.

Although for the sake of clarity we have talked about raising the pump bodily in order to establish various suction lifts, the various changes specified could perfectly well be reproduced by using the test-rig illustrated in Fig. 115 (ii), § 175. Even a throttle-valve on the suction pipe would serve, if it were located sufficiently far away from the pump suction flange.

247. Minimum Pressure in the Pump Passages. In order to study a new aspect of pump performance we can advantageously try a new approach to the problem of representing pressure changes in the pump passages. A commonly-accepted convention for plotting pressure-heads has hitherto been highly serviceable ; it employs positive or negative values as read off directly from suitably graduated pressure or vacuum gauges. Moreover, the same dimensions or the same numerals serve alike for representing pressure-heads and energy per unit weight. From our present viewpoint, though, the system has the defect that the pressure of the atmosphere is itself not shown at all. On the contrary, it is the corresponding pressure-head which serves as the datum or zero to which all other pressure-heads are related. This fact is a useful commentary on our unquestioned belief so far that the pump behaviour is wholly independent of atmospheric phenomena. Of course in the face of the evidence provided by diagrams such as Figs. 173 to 175 we have to abandon this belief ; we are now deeply interested in the atmospheric pressure, for we realise that there is nothing else that will lift the liquid into the pump for us.

The modified system, then, is this : to plot *absolute* pressures instead of positive or negative pressure-heads. Since $p = wh$, this merely amounts, in general terms, to altering the vertical scales of the pressure diagrams such as Figs 109 and 121. More particularly, we must examine still more critically the pressure changes suggested quite schematically in Fig. 121. Since in that diagram we were chiefly interested in energy changes,

§ 194, the actual pressures in the rotor passages were deliberately misrepresented. Another question arises, too. The basic purpose of these pressure-diagrams is to indicate the changes imposed on an element of liquid as it traverses the pump and the pipe circuit. We now have to ask : *which element?* An element that moves along the back of the rotor blade, or an element that moves along the front of the blade ? It was shown in §§ 20 and 21, Figs. 13 and 15, that the element undergoes different experiences according to the path it chooses. These alternative paths must be shown on our new diagram.

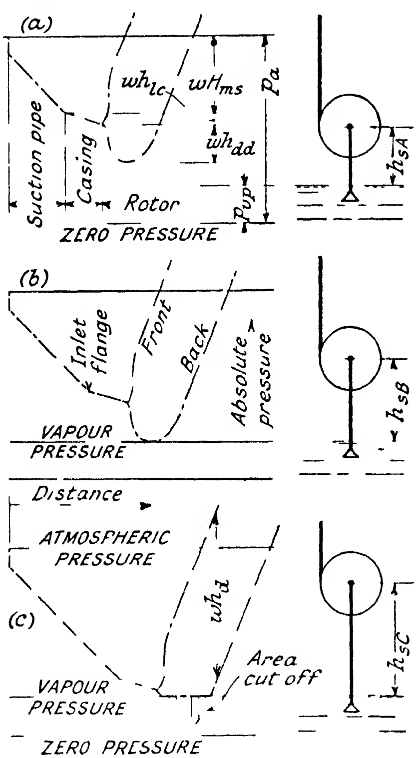


FIG. 176. Pressure-distribution in pumping system before and after cavitation begins.

When we look at such a reconstructed pressure-diagram, Fig. 176, it is seen to include what is perhaps the most important of all the new controlling factors, viz., *the vapour pressure of the liquid* p_{vp} at the temperature prevailing inside the pump. The control is quite firm and rigid. Without any qualification whatever, we have to realise that it is impossible to create in the pump an absolute pressure *lower* than the liquid vapour pressure. Any attempt to do so would merely make the liquid boil : bubbles of vapour would continuously form, but the pressure would not get any lower.

To understand the new convention used in Fig. 176, we may begin at the extreme left-hand of the diagram. Here, before the liquid enters the suction pipe, its *absolute* pressure

is p_a, viz., atmospheric pressure. In passing through the suction system, there is a pressure-drop of wH_{ms} (as in Fig. 109); in traversing the inlet passage of the pump casing, there is a further drop of wh_{lc} (as in Fig. 121). A liquid element choosing to follow the back of a rotor blade next undergoes a final pressure-drop of wh_{dd} (as in Fig. 15 (i)). It is this lowermost part of the pressure-diagram that we shall have to scrutinise most carefully. Thereafter the pressure rapidly rises as the liquid profits by the energy it receives from the rotor blades. As in Fig. 13, the pressure-difference across a blade is represented by wh_d.

248. Causes of Cavitation. If we next proceed to link together our two new forms of diagrams, i.e., Figs. 173 and 176, this is what we find: that the effect of *raising* the pump above the suction well is to *lower* the pressure-diagram relative to the line of absolute zero pressure. To reflect the changes in elevation shown in Fig. 173, then, we can plot in Fig. 176 the changes in internal pump pressures. It will be convenient to assume that the pump speed and discharge are kept steady, leaving the total effective head at the mercy of suction conditions. Diagram (*a*) shows normal operation: the minimum absolute pressure anywhere on the back surface of the rotor blade is well above the limiting value p_{vp}. Diagram (*b*) shows critical conditions: the pump has now been raised sufficiently high to lower the pressure diagram into a critical position: the bottom of the diagram is just touching the line of minimum pressure. Evidently the suction head h_{sB} has attained the critical value, § 246. Meanwhile the general behaviour of the installation has not suffered in any way; the pump still generates its original effective head and shows no sign of discomfort.

There is nothing to stop us from lifting the pump still higher, to the position (*c*), which gives a suction head h_{sC}; but there is certainly something to prevent the minimum pressure from falling any lower. There is the unalterable limit imposed by the liquid vapour pressure. The consequence is that the pressure-diagram is now distorted, Fig. 176 (*c*). The lowermost part of it has been cut off. What is the result? From §§ 20 to 22 we have learned that in comparable conditions the energy impressed on the liquid is proportional to the area of the energy

diagram. If diagram 176 (*c*) has a smaller area than diagrams (*b*) or (*a*), we must conclude that the effect of exceeding the critical suction head has been to reduce the energy given to the liquid, or in other words to *reduce the effective head* H_e. That is exactly what the pressure-gauges show: the manometric head *has* been reduced, § 245.

In the abnormal state of working that we have now reached, certain areas of the back surfaces of the rotor blades are manifestly exposed to the limiting absolute pressure. Brisk effervescence is going on there; elements of liquid as they flow past are momentarily turned into bubbles of vapour. It is the presence of these cavities in the liquid which gives the name *cavitation* to this whole array of low-pressure phenomena.

249. Effects of Cavitation. The generation of vapour bubbles will almost certainly modify the velocity distribution as well as the pressure distribution in the rotor passages (*). There may be deflection of the flow lines of the sort suggested in Fig. 177 (i), and doubtless the combined effect of all these disturbances will be to distort the pressure-diagram much more seriously than was indicated in Fig. 176 (*c*). Still further raising of the pump will increase the area of disturbance, until finally it has extended completely across the rotor passages, Fig. 177 (ii). Thereafter nothing we can do can alter the rate of flow through the pump; the discharge depends *only* upon conditions in the suction system and at the rotor inlet. The pressure drop from atmospheric pressure at the suction-well surface to vapour pressure in the rotor passages is wholly absorbed in overcoming frictional and eddy resistances and dynamic depression head, and in creating velocity head. In Fig. 173 this stage is represented by the vertical parts of the abnormal pump characteristics.

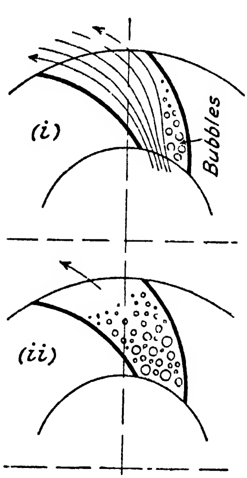


FIG. 177.—Bubble formation in rotor passages

Although, during this culminating phase of cavitation conditions, opening the throttle-valve wider and wider has no effect

on the pump discharge, it is not difficult to see why it induces a steady drop of manometric head. The effect of the vapour bubbles which now permeate a considerable part of the rotor passages is virtually to *lower the density of the liquid.* The pump behaves as if another liquid of less density were flowing through it. Two consequences can be foreseen : (i) as the mean velocity of the fluid mixture is now higher (for the *weight* per second is unchanged), the outlet velocity diagram tends to assume the distorted shapes seen in Fig. 16 (iii) or Fig. 143, which in itself would lower the total head ; (ii) the fall in density reduces the pressure generated, expressed in terms of "solid" or non-gaseous liquid, § 230. For these reasons, then, easing the delivery head on the pump only makes the liquid boil more briskly.

This general picture of cavitation conditions is consistent also with other observed symptoms of irregularity. Audible hissing or roaring is a very normal accompaniment of bubbling ; a lowering in gross efficiency is only to be expected when the flow in the rotor passages is so rudely disturbed. As for the most damaging effect of all—actual physical destruction of the metallic surfaces of rotor and casing—this is described under the heading of effect of length of service, § 257. For the moment the evidence is quite sufficiently convincing : except for very special reasons, the pump must be safeguarded from the manifold dangers of cavitation, and this can best be done by always keeping the suction head within the critical value h_{sc}.

250. Influence of Total Head and of Specific Speed. An answer still remains to be found to the question : why does an increase of total head or an increase of specific speed imply, in comparable conditions, a *reduction* in the critical suction head, § 246 ?

Total Head. By the reasoning given in § 26, it can be shown that when identical pumps are working on their maximum-efficiency curve, § 218, the value of the *relative blade loading* ϵ remains constant. The average *differential pressure-head* h_d is thus directly proportional to the effective head on the pump, H_e. In § 27 it was further implied that the dynamic depression head h_{dd} at least bore some relation to the average differential pressure-head, even if there were no simple mathematical connection between the two. When, therefore, we come to

compare the identical pumps generating different heads depicted in Fig. 174, it seems justifiable to say that as the total head rises, so also does the dynamic depression head(*). Equivalent pressure diagrams are sketched in Fig. 178. Diagram (i) describes the conditions within the low-head pump (left) in Fig. 174; diagram (ii) corresponds with the right-hand pump. The rise in total head from H_I to H_{II} is reflected in an increase in dynamic depression head from wh_{dd} to wh'_{dd}. Since both machines are set at their critical levels, it is clear that the manometric suction head must be eased from H_{ms} to H'_{ms}.

Specific Speed. From the tabular statement given in § 40 we can form a fairly sound impression of the nature of the link between specific speed and relative blade loading. As the specific speed rose from that equivalent to a centrifugal pump to that equivalent to a propeller pump, viz. from 35 to 210 (foot), the ideal value of ϵ increased from 0·73 to 1·95. Applying this knowledge to the two dissimilar pumps of Fig. 175, working under identical heads, we can believe that the pump having the higher specific speed will have the greater average differential pressure-head, the greater dynamic depression head, and therefore the smaller manometric suction head, Fig. 178 (ii).

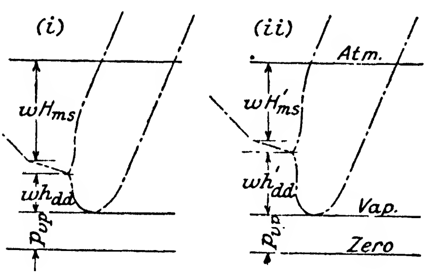


FIG. 178.—Relation between dynamic depression head and suction head.

251. Dynamic Depression Head. As the influence of the dynamic depression head on the suction capabilities of the pump has now been made abundantly clear, it may be worth while to examine other aspects of the relationship. In real or in ideal pumps, we have found out thus far that the value of this depression head h_{dd} may depend upon :—

> the speed N, the flow velocity Y, the head H, the specific speed N_s, and the blade number n [equation (2-2), § 20.]

Another line of approach is provided by the aerofoil theory of axial-flow pump design, § 38. The pressure distribution over the surface of a single aerofoil as in Fig. 24 (i) is well known;

it can be represented by a diagram such as Fig. 179. Here the positive or *negative* dynamic pressures are denoted by the lengths of the vectors, which are read off against a scale graduated in terms of the "coefficient of dynamic pressure", C_k. The numerical value of the pressure at a given point is then computed from the expression $p_k = C_k \cdot \frac{wU^2}{2g}$.

Now Fig. 24 (ii), taken in conjunction with Fig. 179, suggests that in an actual propeller pump the maximum negative pressure-head near the blade tip, is much of the same order of magnitude as the dynamic depression head; and since the relative velocity v_{r1} here corresponds with the original velocity U of the aerofoil, it seems likely that the dynamic depression head will depend only upon the *relative velocity of liquid* at rotor inlet. Again speaking in general terms, this relative velocity is not very different from the inlet blade velocity v_1, and this in turn is linked with the outlet rim velocity v_2. The value of the speed ratio ϕ already gives us a measure of the outlet rim velocity, expressed in terms of the effective head. We could readily use an analogous "inlet speed ratio" to express the value of the inlet rim velocity v_1; reference to Fig. 54, § 92, shows that it would vary from about 0·5 at a minimum shape number of 60 to about 2·1 at a maximum shape number of 600, viz.:—

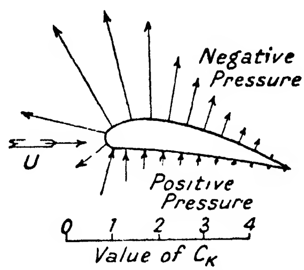


FIG. 179—Pressure distribution on aerofoil

$$v_1/\sqrt{2gH_e} \qquad 0{\cdot}5 \text{ to } 2{\cdot}1.$$

(*Note* that in the axial flow pump, we are concerned with the maximum value of the velocities, i.e., those near the outer rim.)

If we admit that the dynamic depression head does indeed vary as the square of the value of v_1, then it looks as if the *proportional* values of h_{dd} would range from about 0·25 to about 4·2. So here is very strong confirmatory evidence to confirm our belief in the connection between specific speed and critical suction head.

In any event, one general pointer is already clear: to give

a pump the best chance of dealing with a high suction lift, *the inlet edges of the rotor blades must move as slowly as possible.*

252. Summary of Factors Involved. Now that we are within sight of formal rules for assessing the critical suction head, it is as well to be reminded again of the great complexity of the problem (*). The multifarious factors to be taken into account are :—

(i) The atmospheric pressure p_a, which in turn depends upon :—

(ii) The altitude of the pump above sea-level. (Diurnal variations of barometric pressure can be neglected.)

(iii) *Also* the density w of the liquid.

(iv) The vapour pressure of the liquid, p_{vp}, which depends upon :—

(v) The nature of the liquid, at working temperature, and

(vi) The temperature of the liquid under pumping conditions.

(vii) *Also* the inlet loss h_{in} at entry to the suction system.

(viii) the frictional and eddy losses h_{fls} in the suction system.

(ix) The velocity head $\frac{v_s^2}{2g}$ at the pump suction flange

§ 162.

(x) The head loss h_{lc} in the inlet passage of the pump casing, § 194.

(xi) The dynamic depression head h_{dd}, § 251.

Nevertheless when once essential preliminary computations have been made, in terms of *head of liquid at working temperature*, all the variables can be collected into three groups, thus :

(I) Available head-difference $= H_t = h_a - h_{vp} = \frac{p_a - p_{vp}}{w}$.

(II) External head drop $= H_{ext} = h_{in} + h_{fls} + \frac{v_s^2}{2g}$.

(This term includes all changes of head in the suction system, for which the pump itself can be held in no way responsible.)

(III) Internal head drop $= H_{int} = h_{lc} + h_{dd}$.

(This term includes changes of head inside the pump itself.)

The *available head-difference* is all we have to count upon for lifting the liquid through a height h_{sc}, and for overcoming external and internal pressure-drops. This statement can be expressed in the form :—

Critical static suction head $h_{sc} = H_t - H_{ext} - H_{int}$. **(16-1)**

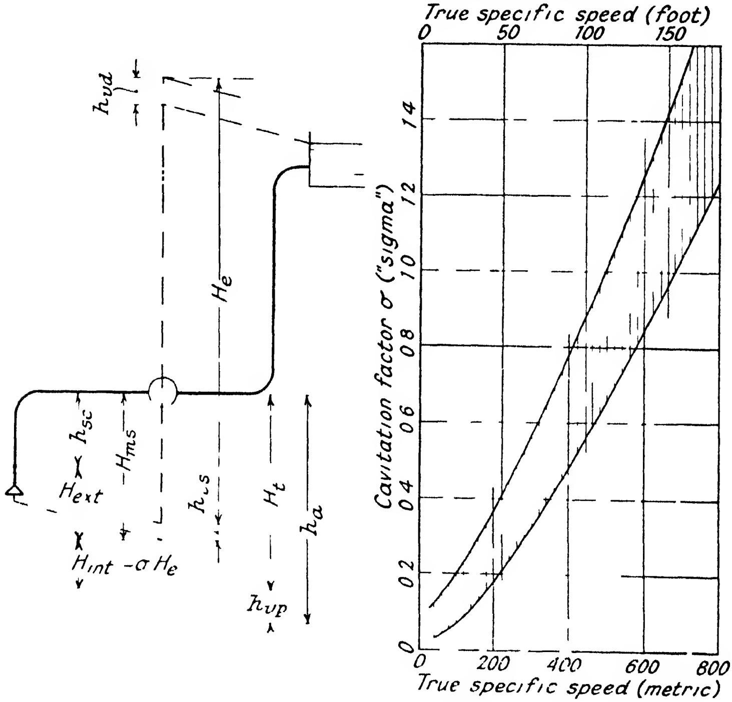


FIG. 180. Connection between specific speed, cavitation factor, and suction lift

Since *manometric* suction head is the sum of static suction lift and external head drop, viz.: $H_{ms} = h_{sc} + H_{ext}$, § 163, a useful alternative form of equation (16-1) is

Critical manometric suction head $= H_{ms} = H_t - H_{int}$. **(16-2)**

The only unknown factor in these expressions is the internal head drop. Experience shows that the trends disclosed by analysis, § 251, are indeed justified, and that the combined term $H_{int} = h_{lc} + h_{dd}$ depends *only upon the total effective head* H_e and upon the *specific speed* N_s. The relationship is expressed with the help of the *Thoma cavitation factor* σ, thus :—

$$H_{int} = \sigma H_e \qquad . \quad . \quad . \quad \textbf{(16-3)}$$

The empirical connection between specific speed and cavitation factor is plotted in Fig. 180.

253. Practical Rules for Estimating Suction Lift. A simple routine is now available for estimating, in given circumstances, the maximum height above the suction well at which it is safe to set the pump. From the known specific speed, the value of the cavitation factor σ is read off from Fig. 180. Then, from equation (16-3), the value of the internal head loss H_{int} is computed. Finally the desired value of the suction lift h_{sc} is found from equation (16-1). The skeleton piping lay-out in Fig. 180 serves as a graphical summary of the process.

Particular weight must here be given to earlier warnings concerning relationships based on specific speed. Although *in a general sense* a given specific speed is associated with a particular rotor shape, yet for that specific speed individual shape ratios may vary through a considerable range, §§ 59, 196. For example, a standard single-inlet impeller will have a bigger diameter ratio d_1/d_2 than an equivalent double-inlet impeller of the same shape number, Fig. 54. The " inlet speed ratio " will thus probably be greater, § 251, from which we deduce that it would be wise to choose a rather higher value of the cavitation factor σ for the one pump than for the other That is why in Fig. 180 a belt or zone of values is plotted, rather than a single line. For *single-inlet* pumps, suitable values of σ are likely to lie in the upper part of the belt ; for *double-inlet* pumps, in the lower part of the zone. In any event, we should not take the risk of setting the pump at its maximum or critical height if it could be avoided. The word " critical " has few reassuring connotations, so if the conditions of installations will permit the pump to be lowered a foot or two, we may very willingly do so. **(Example 33)**

254. Application of Rules. (*a*) At least the suggested rules are in accord with the observed facts stated in § 245. The way in which they control particular installation conditions is shown in Fig. 181. Pumps of various specific speeds are here seen at work, each lifting a stated discharge against a stated static head. As before, pipe friction can for comparative purposes be neglected. Using the *mean* values of σ taken from Fig. 180, the limiting suction lift is found to have the following values :—

Ref Fig 181	Type of Pump.	Specific Speed (foot)	Speed (r p m)	Static head (ft)	Discharge (g p m)	Static suction Lift (ft)
i	Slow-speed centrifugal.	18	480	40	1150	27
ii	High-speed centrifugal.	42	1120	40	1150	20
iii	Screw . . .	80	2200	40	1150	8
iv	Propeller . . .	112	3000	40	1150	1

(*b*) It is to be noted that the information given in Fig. 180 relates *only to pumps working at their respective design points.*

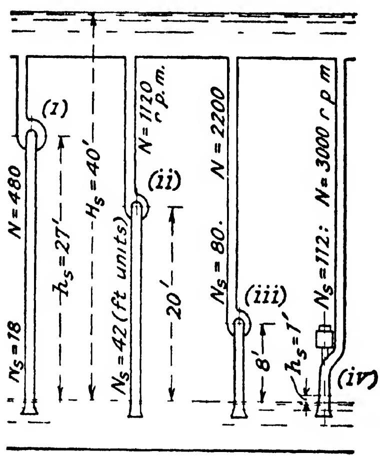


FIG 181.—Influence of type of pump on suction lift, for stated discharge (1150 g p.m) and total head (40 ft.).

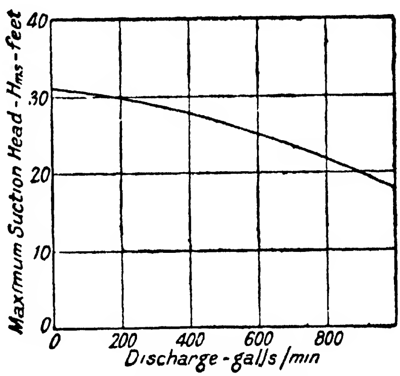


FIG. 182.—"Suction" characteristic for centrifugal pump.

If the discharge is less than normal—if reduced-flow conditions prevail -the suction head may be increased beyond the critical value h_{sc} ; if the percentage flow is greater than 100, the suction head must be diminished. The relation between discharge and permissible manometric suction head H_{ms} is illustrated in Fig. 182 ; the graph relates to a constant-speed pump which at its design point delivers 600 gall./min. against a total effective head of 100 ft. Yet it might not always be judicious to run a pump *continuously* at part-flow against a high suction lift ; the unfavourable conditions at rotor entry, § 204, may in themselves tend to promote cavitation (*).

(*c*) The *size* of the pump has an influence upon the permissible suction head that cannot be shown in diagrams such as Fig. 180. For a stated specific speed it is found that small pumps cannot work against such severe suction conditions as large ones ; this means that for the small machine *a higher value of the cavitation factor σ must be chosen* than for the large machine. If the size of the pump branches is as small as 1 or 2 in., then in any event the suction head should not be more than perhaps 5 or 10 ft. Probably one may find the reason for this divergency of behaviour in the divergency of shape between the large and the small pump. Although *nominally* of the same proportions, yet because of manufacturing necessities the two rotors are not truly geometrically similar ; some possible discrepancies were mentioned in § 227.

(*d*) There is a clear distinction between the *empirical* relationship embodied in formula (16·3), § 252, and the more or less simplified one examined in §§ 245, 246. Although a pump working at points C', D', of its characteristic, Fig 173, might give no audible or visible signs of discomfort, yet " cavitation corrosion " may already have set in, § 257 We must therefore recognise two distinct types of deterioration in pump performance, (*a*) an immediate and unmistakable drop in head and efficiency, § 245, (*b*) a slow decline that only becomes noticeable after months or years of service, § 258. Formulæ of the type of (16-3) are intended to give protection against both dangers.

255. Some Exceptional Conditions. Thus far we have been concerned only with standard types of pump mounted in a standard fashion, viz., *above* the surface of the open suction well from which they drew their supply of liquid Conditions that may require particular examination include .—

(i) "*Negative*" *Suction Head.* If the total head on the pump is sufficiently high, the rules of § 253 may ordain a static suction lift that has a negative value. This means that the pump must be set *below* the level of the suction water surface ; the pump suction branch will be subjected to a *positive* pressure. An installation so arranged is seen in Fig. 227, § 336 (iii).

(ii) *Pump Drawing from Closed System.* Should the pump suction be connected to a closed container or a closed piping

system, there will now be no free liquid surface exposed to the atmosphere which can control the manometric suction head, e.g., Fig. 110, § 163. It will thus be essential to regulate in some other way the conditions within the system, to ensure that the absolute pressure at the suction branch does not fall below safe limits, § 345 (v).

(iii) *Special Construction of Pump.* Here the particular problem is this: suppose that by the rules applicable to normal pumps, § 253, it appears that the suction head must not exceed a certain figure. Yet the conditions of installation may dictate a greater suction head; they make it impracticable to set the pump as close to the suction level as the rules require. What is to be done then? There are at least three possibilities:

(*a*) *Rotor of Special Material.* If it seems that the abnormal suction head to be imposed on the pump will not directly impair its performance or efficiency, then we may be prepared to accept the risk of cavitation corrosion, § 257, provided that the rotor can be made sufficiently resistant. This could be done by retaining a normal shape of rotor, and choosing a special metal for it, e.g., hard bronze, stainless steel, monel metal, nickel alloy, etc., etc.

(*b*) *Pump of Special Design.* Relatively minor changes in the shape of the pump passages may enable the machine to deal successfully with suction heads beyond the range of normal pumps. Thus, generously-proportioned inlet volutes may reduce the head loss in the casing, § 89 (i); or exceptionally skilled design of the rotor blades at inlet may ensure immunity from cavitation. But for certain services – e.g., for condensate-extraction pumps, Fig. 97, § 145—a much more radical revision is indispensable. In order to secure the utmost possible suction capacity, it is essential to give the dynamic depression head its *lowest* value, § 251, while still keeping the impeller speed high enough to generate the specified total head. This implies that the diameter ratio, the flow ratio, and the width ratio of the impeller must all be abnormally low: the resulting effect on the impeller shape is clearly shown in Fig. 97. The effect on the pump *efficiency* can be deduced from §§ 196 to 200, and from Figs. 53 and 54. Because the shape number of the modified impeller is now so far below the normal range, its

efficiency is necessarily sub-normal. But if this particular solution of the problem is chosen, such a sacrifice of efficiency is inescapable.

(*c*) *Rotors in Series.* Let it be stipulated that the abnormally high suction lift must be maintained, and that no decline in efficiency can be tolerated. If we are allowed to use *more than one rotor*, then a standard rotor shape will still serve ; the head generated by the first stage rotor will then amount at the most to one-half of the total effective head, and consequently the critical suction head for the same value of the cavitation factor is substantially increased, § 253, or Fig. 174. According to circumstances, the resulting combination may take the form of (I) a standard type of multi-stage pump, Chapter IX, (II) a special design of multi-stage pump embodying a non-standard first stage rotor, (III) separate pumps disposed in series, § 321 (ii).

Effect of Age and Length of Service

256. Types of Deterioration. In a closed iron passage carrying liquid, we may expect that after some time the walls will have become rusted or corroded or in some way roughened. The passages of the rotor and casing of a rotodynamic pump are liable to just the same type of deterioration ; but in addition they may be subject to much more destructive attacks—the attacks of *cavitation corrosion.* Leaving this special problem for later study, § 257, we can here enumerate a few examples of normal wear in a pump :—

(i) The rate of corrosion of the pump parts will naturally depend upon the nature of the liquid and upon the nature of the materials of construction. Frequent references, §§ 140, 147, have shown how the liquid and the material can be matched one against the other. Evidently a cheaply-built pump, in which cast iron is used for the main components, may become unserviceable sooner than a pump fitted with a bronze impeller. In any event the effect of corrosion will be to increase the hydraulic losses in the pump.

(ii) The running clearance between impeller eye and pump casing will increase, § 82. The rate of wear will be stimulated if the liquid is dirty or gritty, or if the pump shaft is allowed to run out of truth.

(iii) Wear on the shaft, or on the shaft sleeves, § 80, or on the neck-bushes, § 86, may likewise increase leakage.

(iv) In multi-stage pumps fitted with balance-discs, wear on the disc may slightly displace the whole rotating element endwise, thus throwing the impellers a little out of alignment with their diffuser rings, § 75 (*c*). The running clearance between shaft sleeves and inter-stage diaphragms, § 122, may gradually increase.

Here are inevitable causes for deterioration of performance, quite apart from defective maintenance of the glands and stuffing-boxes.

257. Cavitation Corrosion. This is the name given to the destruction of metallic surfaces that nearly always occurs if a pump is allowed to run for long periods at a suction head substantially greater than the critical value, § 249. In these conditions we know that bubbles of vapour are generated in the zones of minimum absolute pressure, and that they collapse again as soon as the natural flow through the rotor sweeps them into regions of higher pressure. What makes these tiny bubbles so dangerous is that their rate of collapse is extremely rapid. The liquid hastening to fill up the vacuous spuces left by the bubbles is almost instantaneously arrested by the metallic blade surfaces, thus engendering serious inertia pressure or *water hammer.* That is to say, as each bubble in contact with the blade surface breaks down, a very tiny area of the surface is momentarily exposed to an extremely heavy stress. The effect is almost as if the metal surface were undergoing bombardment by myriads of very minute hard pellets.

The phenomenon (which is alternatively termed " cavitation erosion ") is as yet only imperfectly understood; it is possible that chemical corrosion due to the liberation of oxygen from the liquid is also responsible; but the results are quite unmistakable. Metal surfaces are ravaged as though gnawed by rodents; solid cast iron an inch or two thick may be eaten clean through. The examples of damaged rotors illustrated in Figs. 183, 184, give a good impression of the severity of the effect. At least in the propeller-pump blade, Fig. 184, the cavitation corrosion is located just where one would expect—near the leading edge and close to the outer rim where velocities are highest, § 36. The distribution of damage over the

centrifugal pump impeller, Fig. 183, is not so easy to explain; and indeed one of the perplexing features of cavitation corrosion is that it may attack quite unexpected areas of the rotor and casing.

So-called " edge corrosion " may occur at the blade edges of open-type rotors of all kinds, § 69 (III), propeller-pump blades being peculiarly liable to this kind of damage. In such pumps the differential head between front and back of the blades is at its maximum in relation to the total effective head, § 40; and this differential head generates a correspondingly rapid

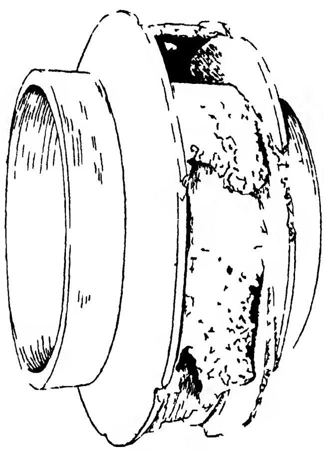


Fig. 183 —Result of cavitation erosion on centrifugal pump impeller

Fig. 184 Propeller pump rotor blade damaged by cavitation erosion

leakage flow across the clearance between circumferential blade edge and the fixed casing. As a result of the eddying on the downstream side of the clearance space, there is an abnormal local pressure-reduction which may be great enough to set up local cavitation. If so, the blade edges will be attacked, and in time the leakage loss will substantially increase.

It is to be remembered that the mechanical vibration that accompanies severe cavitation may accelerate wear on the pump bearings, especially if uneven erosion of the rotor blades has already thrown the rotating element out of balance.

258. Effect of Wear on Performance. Detailed analysis of power losses as developed in § 219 might indicate how length

of service influences a pump's characteristic performance. But just as in § 225 an alternative approach was chosen, so now it may be illuminating to try still a third method. It depends on the notion of separating the leakage flow from the effective flow through the pump, § 191. In comparing the pump when new with the pump when old, let us assume that the *speed* remains unchanged and that the *total* flow through the rotor passages, $Q + q_l$, is likewise unaltered. This implies that the energy input to the rotor has not been substantially affected, nor has the shaft horse-power input. The only basic change is in the rate of leakage—for we can study later on the effect of corrosion. As a result of wear of all kinds, let the total leakage loss be increased from q_{l1} to q_{l2}. This means that the *effective* flow through the pump delivery branch has *fallen* from Q_1 to Q_2, where $Q_1 + q_{l1} - Q_2 + q_{l2}$. The drop in effective discharge, $Q_1 - Q_2$, may be denoted by dQ.

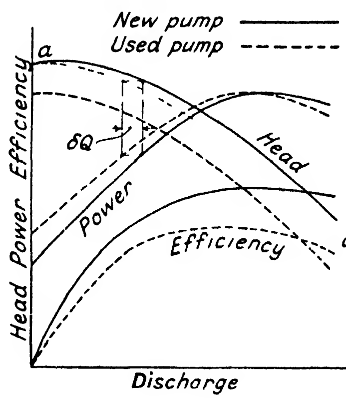


FIG 185.—Characteristics for new and for old centrifugal pump.

Comparing now the "new" with the "worn" characteristics, we see that for a given total or rotor flow the head and power ordinates are unaltered in height: they are merely shifted by a distance dQ towards the origin of the graphs, Fig. 185, because the effective pump discharge has fallen by that amount. At this stage, the provisional head discharge curve is indicated by the light broken line, *aa*. Finally the influence of the roughening of the passages due to corrosion can be estimated. Since it will undoubtedly increase the hydraulic loss H_L, § 195, the effective head must fall, and with this knowledge we are justified in sketching in the final head-discharge characteristic, Fig. 185. It only remains to compute, from the accepted head and power characteristics, the efficiency curve for the worn pump, and then we have before us a complete picture of the changes that the pump has suffered during service.

In general, for a stipulated effective rate of discharge, the *head* has fallen, the *efficiency* has fallen still more, and the

power input has *risen*. For very small or very large flows, the effects are less easy to predict: they depend a good deal upon the shape of the original characteristics. If cavitation erosion has been tolerated, naturally the behaviour of the used pump is correspondingly worse.

259. Other Aspects of Impaired Behaviour. A correct estimate of the rate at which the pump performance will decline may assist both in the original design of the pump and in the framing of a maintenance programme, § 361. Clearly if the stipulated design conditions are to be strictly enforced throughout the life of the pump, they must be capable of fulfilment not only by the new machine but by the *used* machine. When the pump leaves the makers' works, that is to say, it must have a reserve of performance in hand. The question of how much reserve may be influenced by the conditions of installation; for instance, in the conditions laid down in § 240 (i), quite a slight displacement of the head-discharge curve would reduce the discharge below permissible limits, but such a displacement would have much less serious consequences in the conditions of § 240 (ii).

At some time or other in the life of the pumping installation, one may foresee that the decline in performance will transgress reasonable limits; either the minimum required head or discharge can no longer be maintained, or else running costs have become exorbitant. Remedial measures include: (i) replacing the complete pump, (ii) replacing the rotor and shaft, (iii) replacing wearing rings, neck-bushes, shaft sleeves, etc., making good eroded areas by welding, etc. For very small pumps the first method is likely to be the most economical one; for very large ones, the third. The question is discussed in greater detail in §§ 298, 360-362.

CHAPTER XVII

TRANSITORY CONDITIONS

260. Phenomena during Starting and Stopping. The study of pump performance that has been carried through the preceding chapters appears to be valid only when the pump is running at uniform speed and the liquid is flowing uniformly through it. But this restriction cannot be maintained in practice. The pump must often be started and stopped; the rate of flow may vary from moment to moment. A comparison with a moving vehicle may give us an inkling of what to expect during periods of acceleration or retardation. When, for example, the vehicle is brought to rest by violent application of the brakes we have vivid personal sensations to remind us of the severe stresses that may be imposed on the parts of the vehicle. The analogy is a just one in this sense, that we do in fact find that a pumping system may be subjected to quite dangerous pressures during acceleration and retardation of the liquid column in the pipes. Only a detailed analysis of the complex conditions that may develop will show how to protect the pump and pipes, and how to frame a safe routine of starting and stopping.

The energy which, if not controlled, may wreck the system is of two kinds, (i) the kinetic energy of the moving column of liquid, and (ii) the elastic energy of the liquid and of the pipes. In the same way that energy may be stored in a steel spring, so energy can be stored by compressing a liquid or by expanding the metallic walls which contain it. The problem during the transitory conditions now to be examined consists in applying

this energy or dispersing it without violent consequences. It seems likely that the most severe conditions will correspond to instantaneous starting and stopping of the pump; and although such conditions will only be imaginary they will form a simplified guide for the relatively gradual changes usually found in a pumping circuit.

261. Instantaneous Starting. The pump has a short vertical suction pipe with foot-valve, § 290, and delivers through a horizontal pipe into a reservoir, Fig. 186 (i). When liquid

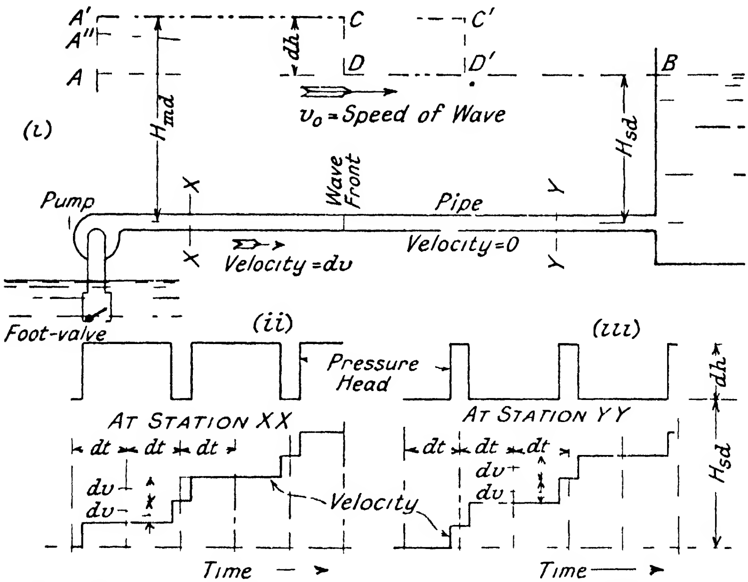


FIG 186 Pressure and velocity changes after pump has been started instantaneously

and rotor are at rest, the pressure-line or hydraulic gradient will be horizontal, AB. Starting the pump in (assumed) zero time will instantly raise the pressure head at the delivery flange from H_{sd} to H_{md}, the new value being controlled by the pump's characteristic, § 212. At the same instant the liquid at the delivery flange will begin to move with velocity dv. Meantime the liquid in the rest of the delivery pipe *remains inert*. But it very quickly comes to life. The increased pressure-head generated by the pump now sweeps along the pipe in the form of a wave which is represented by a step in the hydraulic

gradient. After a very small interval of time the step has advanced to position CD; after another interval, to $C'D'$, and so on. By the time the wave reaches the open end of the delivery pipe, the whole of the liquid has been subjected to the pressure-increment dh and it is all moving with velocity dv. The wave now reverses its direction, and returns without any change of speed back to the pump again, the liquid column acquiring meantime a further increase of velocity. The resulting changes in pressure-head and in velocity, as observed for instance at two stations XX and YY, are plotted in the graphs (ii) and (iii), Fig. 186. The variables are here plotted on a time basis, the symbol dt denoting the time required for the pressure-wave to make *one* trip from end to end of the pipe.

A remarkable simplicity characterises the relationship between the factors :—

v_0 — speed of travel of pressure wave,
dv — increment of velocity corresponding to an increment of pressure head dh.

It is :—

$$dh = v_0 \cdot \frac{dv}{g} \qquad . \quad . \quad . \quad (17\text{-}1)$$

For a given pipe and a given liquid, the speed of the pressure-wave v_0 is *invariable*; it has the value

$$v_0 = \sqrt{\frac{g}{w\left(\frac{1}{K} + \frac{D}{TE}\right)}} \qquad . \quad . \quad . \quad (17\text{-}2)$$

where D = internal diameter of pipe,
T = thickness of pipe wall,
E = Young's Modulus for the pipe material,
K = Bulk Modulus for the liquid.

If the pipe walls were rigid and inextensible, then the speed of the pressure wave would be identical with the velocity of sound in the liquid—about 4700 ft./sec. for water. In fact, the velocity is a little less—perhaps 4000 ft./sec. or so.

262. Accelerating the Liquid. Although just now we assumed that each element of liquid in the delivery pipe receives its acceleration in the form of a series of jerks or momentary impulses, yet the mean velocity of the whole column rises quite

smoothly and regularly. This *mean acceleration* has the value :—

$$\frac{dv}{dt} = g \cdot \frac{dh}{l},$$

viz. it is equivalent to the acceleration of gravity multiplied by the ratio of the pressure-head increment to the pipe length l.

While the liquid column is gaining speed, the pressure-wave ceaselessly sweeps to and fro along the pipe with unchanging velocity v_0. Meantime the flow through the pump has now increased so materially that the head generated at constant rotational speed may begin to change as dictated by the pump characteristic. Furthermore, the velocity through the pipe is high enough to generate an appreciable frictional loss. The combined result is that the rate of acceleration progressively

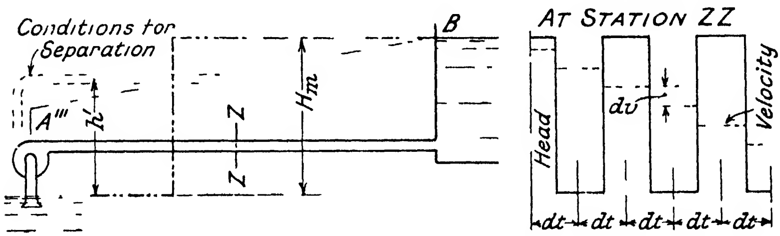


FIG. 187. - Pressure-fluctuations after instantaneous stopping of pump.

declines. In the end, after a number of seconds or maybe several minutes from the moment of first switching in the pump, acceleration ceases altogether ; the pressure-waves have been completely damped out, and the whole system has settled down to normal working routine. The slope of the hydraulic gradient $A''B$ in Fig. 186 (i) is accounted for entirely by frictional loss, and thus the steady conditions resemble those of § 240 (ii).

263. Instantaneous Stopping. It will be convenient first to assume that the suction pipe of the system has no foot-valve, Fig. 187, and that the pump is running normally. The sudden stoppage of the pump sets in motion a pressure-wave of the same nature as before ; but its amplitude is now much greater —it is in fact equivalent to the manometric head on the pump. The mean rate of retardation of the liquid column is correspondingly high. The column quickly comes completely to rest, and then it reverses its direction and begins to travel

backwards; liquid drains back from the reservoir into the suction well. In consequence the pump rotor begins to revolve backwards-way also; it behaves like a turbine runner. Although these reversals have no effect on the pressure-wave, which continues to course to and fro with unimpaired speed, yet before long, perceptible damping occurs; in the end, all pressure-fluctuations subside, and the final flow picture can be represented by the line $A'''B$, Fig. 187. The rate of uniform return flow through the system will depend upon the frictional resistance of the pipe and upon the hydraulic resistance of the pump rotor, whose terminal reversed speed may be much higher than its normal forward speed.

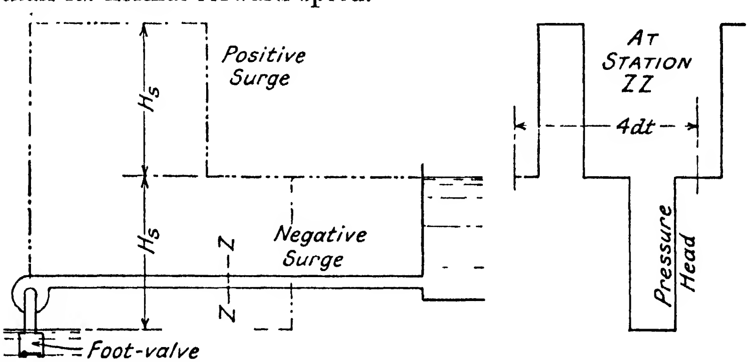


FIG. 188.—Positive and negative pressure surges after closure of reflux-valve.

Effect of Foot-valve. The presence of an automatic foot-valve or other form of reflux-valve in the circuit, Fig. 188, has no appreciable effect until the liquid column is on the point of coming to rest. When the liquid now tries to run backwards it is brusquely checked by the foot-valve, which has just closed. The resulting inertia pressure or *water-hammer* converts the wave of negative pressure (relative to the delivery level) into a positive pressure whose intensity may be as great as the original static head; that is to say, the maximum pressure on the suction pipe is *double* the static head. It is this positive wave that now traverses the system, Fig. 188. Another negative surge will follow, and the alternation will continue until the available energy in the pipe at the moment of foot-valve closure has been wholly dissipated. During this period, of course, the liquid column *as a whole* is stationary.

264. Separation in the Liquid Column. Figs. 187 and 188 show an important secondary effect of instantaneously stopping the pump. As soon as the step in the negative pressure wave has traversed a given point in the delivery pipe, the pressure-head at that point changes from positive to negative (using these terms in their *usual* sense); viz. the pressure falls to sub-atmospheric. What would happen if the delivery pipe were to be raised to the new position, Fig. 187 (broken lines), everything else remaining as it was? If the vertical height h' were 50 ft. or so, apparently a negative head of 50 ft. would be generated. But we know that a negative head of 34 ft. of water is the utmost that can by any possibility be realised. The net result of these conflicting influences is what is termed *separation*. At the point in the pipe at which the limiting negative head is first attained, the liquid column breaks, and an empty vacuous space is formed between the two portions. As the rear part of the column suffers far more violent retardation than the front part, the intervening space at first widens; but later on, the gap closes The impact between the two parts of the column when they re-unite is quite exceptionally severe, and would almost certainly cause damage to the line.

265. Actual Conditions during Starting and Stopping. Turning now from imaginary and artificial conditions to actual operating procedure, it is evident that the general characteristics of the pumping set as a whole may have a pronounced influence on pressure changes in the line. The moment of inertia of the rotating parts, and the energy required to accelerate them, must certainly be taken into account During the accelerating period the motive unit—engine or motor—is called upon to deliver energy not only to the liquid column but also to the mechanical elements of the set. Similarly the energy stored in these revolving elements will to some extent control the rate of retardation of the liquid, when the pump is stopped. The nature and position of the valves in the pipe-line must also be studied; for example, reflux-valves or sluice-valves. In regard to the pump itself, we are required to make this fundamental assumption: that although the characteristic curves supplied to us were derived from observations taken each with the pump running at a *steady* speed and delivering a *steady* flow, yet we

must accept them as being applicable also during rapidly-changing conditions. During the transitory regime now being studied, we know that the pump speed will only instantaneously have the value N; nevertheless if at that instant the rate of discharge has the momentary value Q, then we shall claim that the corresponding instantaneous head H has the value read off from the appropriate characteristic.

The sum total of factors that may have to be taken into account during variations in flow conditions thus include :—

(i) Length, diameter, wall thickness, and lay-out of piping.
(ii) Position and nature of valves.
(iii) Nature of liquid.
(iv) Initial mean velocity of liquid.
(v) Head-discharge and power-discharge pump characteristics.
(vi) Characteristics of starting-mechanism of pumping set.
(vii) Relation between speed and kinetic energy of revolving parts.

266. Starting the Pump. (i) *Against Closed Throttle.* A common method of starting centrifugal pumps is to keep the delivery sluice-valve or other valve fully closed during the whole accelerating period, § 354. Only when the pump is running steadily at normal speed is the valve gradually opened and the accelerating head applied to the liquid column, which has hitherto remained inert. The pressure waves such as those which would correspond to instantaneous starting, Fig. 186, are now almost imperceptible; the whole operation is entirely under the control of the attendant.

In regard to the torque to be applied to the rotating parts of the pumping set, we know that at the instant of first starting from rest, the pump rotor delivers no energy to the liquid; the gross starting torque is absorbed wholly in accelerating the pump rotor, the couplings, and the moving parts of the motive unit. As the pump gathers speed, the torque demanded by its rotor increases, its value being obtainable from the closed-throttle point of the appropriate characteristic. The difference between this torque, and the total torque applied to the set, gives the residual torque available for generating angular acceleration (*). If an electric driving motor were in question, it would thus be possible to design the starting gear

to suit a stipulated maximum current, or to choose between a squirrel-cage or slip-ring induction motor, etc., etc., § 280.

(ii) *Against Reflux-valve only.* Here the pipe system is assumed to be unobstructed except by a reflux-valve which keeps the liquid column from running backwards, § 290. After the pumping set has been started up, the sequence of events is identical with what occurs under closed-throttle conditions, (i) above, up to the point at which the mounting pressure forces open the valve. Immediately an accelerating force is impressed on the liquid column. A pressure wave will make its way along the pipe; although its amplitude may be much less than was depicted in Fig. 186; and although the wave form will certainly be different, yet the pressure variations may be quite pronounced, especially if a powerful starting torque is available. Such conditions arise if a direct-on line motor starter is used, § 280. Moreover, as flow at an increasing rate occurs through the pump rotor, the equivalent power input also rises, with a corresponding effect on the net torque available for acceleration.

Special problems met with in *propeller-pump* installations are mentioned in § 327. **(Example 34)**

267. Stopping the Pump. Just as it is often good practice to start a pump against closed throttle, so also is it desirable to stop it against closed throttle. With the pump running steadily, the delivery valve is gradually closed until the whole liquid column is " dead " ; then and only then is power cut off from the motor. Alternatively, the speed of the pumping set may gradually be reduced by hand regulation so as to bring the liquid column to rest without major pressure fluctuations. In either case we assume that the operation is wholly under the control of the attendant. But suppose the attendant is inexperienced or careless or absent altogether ? Suppose he shuts down an electrically-driven set by pulling out the main switch before closing the delivery valve ? Or suppose there is a disturbance in the electric power supply— a current failure or the like—that causes the motor to trip out automatically ? These are normal risks that must be accepted, and that call for suitable protective measures. In considering what this protective technique shall be, it will only be possible in this chapter to study the special case of electric pumping sets from which the

supply-current is suddenly cut off, either deliberately or accidentally.

Until the current is broken, the set is running at a steady speed and the whole of the energy delivered to it is either transferred to the liquid or dissipated in mechanical or fluid friction, in heating the surrounding air, etc. The instant after the power supply is interrupted, there remains only one source from which energy can be drawn for feeding to the liquid : it is the kinetic energy stored in the rotating parts of the set—the pump rotor, the couplings, the motor armature or rotor, and so on. It is a strictly limited source. The instant it is exhausted, the pumping set must come to rest. The rate of retardation of the revolving parts at any instant will depend upon the moment of inertia of these parts and also upon the hydraulic conditions at that instant, i.e. upon the speed, head, and discharge of the pump. It can be estimated thus :—

Let I = moment of inertia of whole revolving masses,
= combined weight × (radius of gyration)^2,
δN = drop in speed during a small interval of time δt,
W, H_e, N and η_m have their usual significance (discharge, effective head, speed, and gross efficiency) and their momentary values during this small interval of time,
K_t is a constant, which includes the value of the acceleration of gravity, g.

Then it can be shown that

$$\textit{rate of retardation} = \frac{\delta N}{\delta t} = \frac{K_t W H_e}{\eta_m N I} \quad . \quad \textbf{(17-3)}$$

If I is expressed in lb. weight (feet)^2, W in lb./sec., H_e in feet head, and N in revs./min., then the constant K_t has the value 2950.

If I is expressed in kg. (metres)^2, W in kg./sec., and H_e in metres head, then K_t = 936.

268. Shape of Pressure-wave. By combining a step-by-step method with a system of trial and error, we can now use equation (17-3) to plot curves between time after interruption of power supply, and (*a*) speed of set, (*b*) head generated. They are seen in Fig. 189 (i) ; the slope of the time-speed curve at any point has the computed value $\delta N/\delta t$. In turn these curves

enable us to construct, for the actual conditions of gradual retardation of the pump, the shape of the pressure-wave that we have already drawn for the imaginary case of instantaneous stoppage, Fig. 187, § 263. We first assume that the gradual slowing down of the pump rotor is carried out in small sudden jerks or steps, as indicated in Fig. 189. If the period of retardation is divided into a number of small equal intervals, then at the mid-point of each interval the head is imagined to drop suddenly, thereafter keeping steady until the next drop.

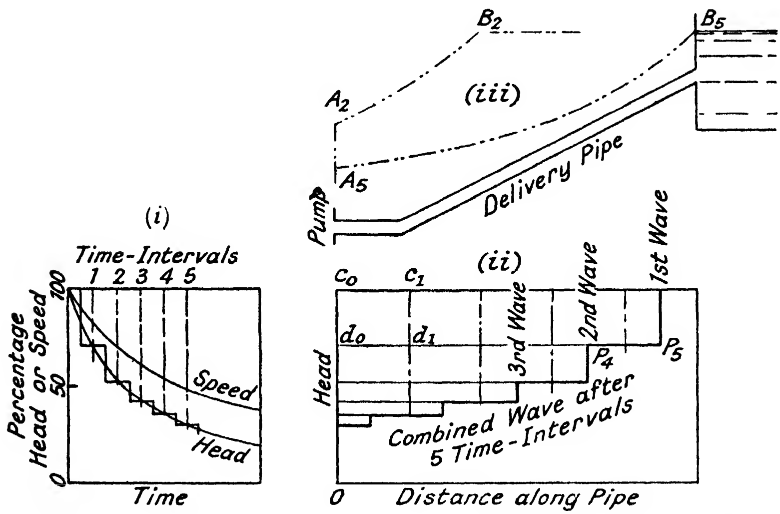


FIG. 189.—Step-by-step plotting of pressure-changes during gradual pump stoppage.

By making the time-intervals sufficiently small, we can approach as closely as we please to the actual smooth curve, diagram (i).

Each sudden change in head will generate a pressure-wave along the pipe of precisely the same rectangular form, and travelling at the same speed, as the wave depicted in Fig. 187 ; but the height of the wave is now much smaller. The true pressure-head in the pipe at any point and at any moment may be found by combining the effects of the individual waves ; and this summation can readily be made graphically as in Fig. 189 (ii). In this diagram the horizontal distance c_0c_1 or d_0d_1 represents the distance traversed by any wave in any one time interval, while the vertical height c_0d_0 represents the

amplitude of the *first* wave. Thus at the end of five time-intervals the first wave of amplitude c_0d_0 has reached point P_5 in the pipe, the second wave has got as far as P_4, and so on. By smoothing out the steps in diagram (ii), we arrive at a fair presentation of the true pressure-wave ; the line A_2B_2, Fig. 189 (iii), shows the distribution of pressure-head after two time-intervals, and the line A_5B_5 shows the state of affairs after five intervals.

Comparing now the general types of waves in Figs. 187 and 189 (iii), we observe that the true wave is much less likely to be dangerous than the imaginary one. If a line such as A_5B_5 in Fig. 189 (iii) represented extreme conditions, then the pump delivery pipe could have the slope shown in the diagram without exposing it to the risk of abnormal pressure-shocks resulting from " separation ", § 264 : at no point would there be a negative head in the pipe. We can admit, then, that this system of representing transitory phenomena is most illuminating. The trouble is, unfortunately, that the method of construction is extremely tedious. The reason lies here : the various diagrams in Fig. 189 convey no information whatever about the *discharge* through the pump when once it has begun to slow down. Yet the relation between pump speed and head plotted in diagram (i) cannot in fact be established until the momentary discharge is known, § 267, these three variables being rigidly linked by the shape of the pump characteristic curves. To eliminate this uncertainty, and the trial and error system it imposes, we may turn to the very elegant graphical method recently developed by Professor R. W. Angus, Professor Bergeron, and others (*). By following the precise instructions now to be given, designers can confidently use the method even if they have no time to study the underlying principles.

269. Graphical Plotting of Transitory Conditions. In these instructions it is assumed that the pump delivers against a static head H_s into a long pipe of uniform diameter and of length l, as in Figs. 186-189. Friction and velocity heads are disregarded. The successive steps are :—

(i) From the given head-discharge pump characteristics, prepare a new chart in which head is plotted against *delivery pipe velocity* v instead of against pump discharge, Q. This can quickly be done by tracing the original curves just as they

stand, merely altering the graduations of the horizontal axis, i.e. dividing discharge by pipe area so as to yield $v = Q/a$, Fig. 190 (i).

(ii) From the known details of the delivery pipe and of the liquid, calculate the speed v_0 of the pressure-wave along the pipe (17-2), § 261, and hence find the time $dt = l/v_0$ required for the wave to make *one* journey along the pipe.

(iii) From equation (17-3), § 267, and the known particulars of the pumping set, calculate the drop in speed $2\delta N_0$ corresponding to a small interval of time $2\,dt$, viz., the period of time needed for the pressure-wave to travel from the pump to the open end of the pipe and *back again*. If N_0 was the original pump speed, then the speed after $2dt$ seconds from the moment of tripping out the motor will be $N_2 = N_0 - 2\delta N_0$.

(iv) By interpolation from the original $H - Q$ characteristic curves, plot the head-velocity curve corresponding to speed N_2, as shown at $N_2 - N_2$ in Fig. 190 (i).

(v) Calculate the numerical value of the expression

$$\frac{v_0}{g} = \frac{\text{speed of pressure-wave}}{\text{acceleration of gravity}},$$

(attending carefully to *units* (17-1), § 261). Henceforth the entire graphical construction consists *only* of a network of straight lines having a slope v_0/g, intersecting a series of head-velocity curves. If the distances AE, AC and AD, Fig. 190 (ii), are set off so that the ratio

$$\frac{\text{units of head plotted vertically}}{\text{units of pipe-velocity plotted horizontally}}$$

has the specified numerical value v_0/g, then the construction lines can at once be drawn parallel either to DE or to DC.

(vi) From the point O representing steady flow conditions before stoppage, draw construction line Oa_2 intersecting characteristic curve $N_2 - N_2$ at point a_2.

(vii) Using now the reduced speed N_2, calculate from equation (17-3), § 267, the pump speed N_4 after a further time interval $2\,dt$. Plot the corresponding curve on the chart.

(viii) Draw construction lines a_2b_2 and b_2a_4, touching the static head line at b_2. As we have agreed to ignore friction head and velocity head, the effective head on the pump is equal to the static head H_s.

(ix) Continue for speeds of N_4, N_6, etc., until the original velocity is exhausted, viz. until the construction lines begin to cross the axis $v = O$.

(x) Now interpolate new head-velocity characteristics relating to speeds $N_1 = N_0 - \delta N_0$, $N_3 = N_2 - \delta N_2$, etc., as shown by broken lines in Fig. 190 (i). (The speed-drop δN_0 corresponds to a time-interval dt, or period for the pressure-wave to make *one* single journey along the pipe.) Add the construction lines shown broken in the diagram.

(xi) Finally join the points $O \ldots a_1 \ldots a_2 \ldots a_3$, etc., by the heavy line as indicated.

270. Interpretation of the Velocity-head Diagram. What information can we extract from the completed diagram,

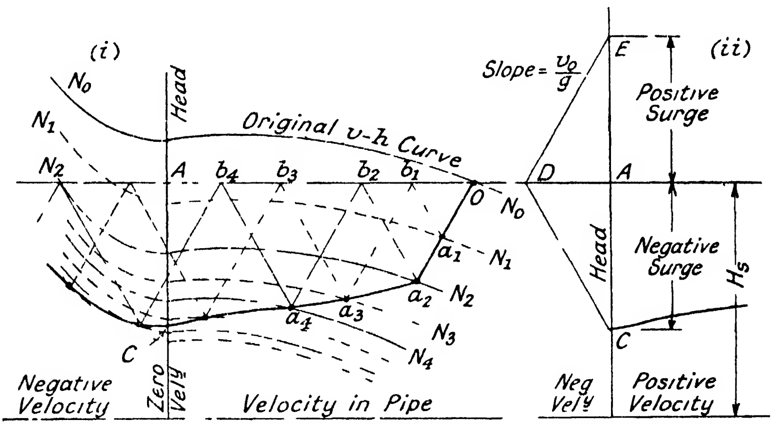


FIG. 190. Graphical construction for evaluating intensity of pressure-surges.

Fig. 190 ? It is as follows : (*a*) The heavy line $O \ldots a_2 \ldots a_4$ is a record of pressure-head variations *as measured across the pump flanges.* It shows at once that by the time the liquid column has come to rest, the pressure-head difference has fallen from the original value H_s to the minimum value $H_s - AC$. (*b*) Although heads are here plotted against pipe velocity, it is an easy matter to transfer them to a time basis, for we know that the successive points O, a_1, a_2, etc., relate to successive time intervals of $dt = \frac{l}{v_0}$. Thus the time-pressure diagram, Fig. 189 (i), which we could hitherto only have constructed very laboriously, can now be plotted straight away. Indeed

if it is our purpose to draw the shape of the wave in Fig. 189 (ii), there is no need to plot diagram (i) at all; by giving the time-intervals between the sudden head-reductions the value dt, then the corresponding reductions of pressure-head can be scaled off directly from Fig. 190 (i). (*c*) Readers who have interested themselves in the principles of the graphical method will observe that the points b_2 and b_4, for instance, represent to scale the velocity in the pipe *at the open end* near the reservoir, after periods (from the moment of tripping out of the motor) of 3 dt and 5 dt respectively. **(Example 35)**

(*Note.*—Although the graphical method is fundamentally flawless, certain corrections may be desirable in putting it into effect. The neglect of friction loss and velocity head in the pipe has already been mentioned. Furthermore, the appropriate values to be inserted in equation (17-3) should really relate to conditions at the middle of the period under construction, not those at the beginning of the period. Again, this equation takes no account of the energy absorbed during the retardation period by the motor bearings and by armature windage. On the other hand, the pump gross efficiency can quite permissibly be interpolated from the iso-efficiency curves, § 222.)

271. Return-flow Conditions. The velocity-head diagram, Fig. 190, has by no means exhausted its utility; it will continue to help us in studying the behaviour of the pumping installation after the liquid column has come to rest. If the system includes *no reflux valve* or the like, then the retardation of the column will be maintained and will lead to a negative velocity in the pipe—the liquid will begin to flow backwards through the pump, which itself may still be running in its normal forwards direction, although with rapidly-falling speed. There are two phases in the ensuing sequence of phenomena: (i) the residue of kinetic energy in the rotating parts will be used up and the set will come to rest, the liquid meantime flowing backwards through the rotor at higher and higher speeds, (ii) the set itself now begins to run backwards-way, and continues to accelerate until the terminal steady conditions sketched in Fig. 187, § 263, are reached, viz. the entire static head is absorbed in generating velocity head and in overcoming the friction of the pipe and the hydraulic resistance of the pump, which itself is now behaving as a turbine. The first phase only will be examined here.

As soon as the construction lines of Fig. 190 (i) overstep the line of zero velocities, they enter an uncharted region; after all, one cannot expect a pump-maker to say what his

machine would do if liquid flows through it backwards-way, nor would it be complimentary to him to suggest that his pump would ever be thus humiliated. But there is little difficulty in setting up a test-rig which will yield the required information ; it will embody an overhead tank or the like for forcing water backwards-way through the pump, at measured speeds and rates of reverse flow (*). When, after such experiments, the normal pump characteristics are extended into the zone of negative flows, they are found to have the form seen in the left-hand side of Fig. 190 (i). By continuing the pattern of construction lines onwards from point b_4, we see that the pressure-head generated at the pump flange now begins to rise slowly from its lowest value at zero discharge.

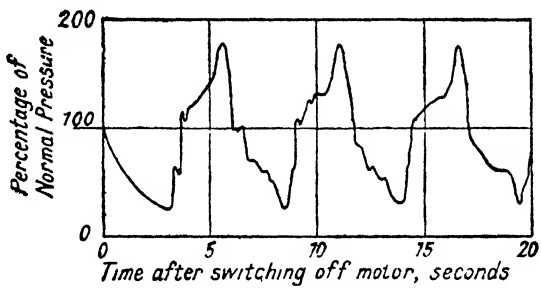


FIG. 191 —Pressure wave in actual pumping installation.

272. Water-hammer created by Reflux-valve. The interposition of a non-return valve in the circuit shown in Fig. 188, § 263, had the effect of instantaneously generating a *positive* pressure surge. A similar result will follow a similar cause in the actual conditions of gradual pump stoppage ; but naturally the intensity or amplitude of the surge may now be less than it was when sudden pump stoppage was stipulated. The maximum rise in pressure-head thus experienced in the pipe near the pump delivery flange can be scaled from the right-hand velocity-head diagram, Fig. 190 (ii). From point C, representing minimum pressure at the moment of zero pipe velocity, the construction lines CD and DE are drawn at their stipulated inclination (if they have not already been drawn), and the point E then represents the desired *maximum* pressure. Just as in Fig. 188, the positive pressure surge has the same numerical value as the negative surge.

An actual autographic time-pressure record traced during the shutting down of a large pumping set is reproduced in Fig. 191. The first part of the curve, from the moment of cutting off power to the moment the reflux-valve closes, closely resembles the computed curve, Fig. 189 (i); thereafter the series of peaks and hollows, corresponding to the ideal rectangular graph in Fig. 188, shows how the wave form is modified by the rate of retardation of the pumping set and by the peculiarities of the pipe-line (*).

273. Slam-pressure due to Uncontrolled Automatic Reflux-valve. Hitherto we have believed either that the ideal foot-valve closes instantly, the moment the liquid column comes to rest, or else that the actual foot valve is so contrived or controlled as to give the same effect. Ordinary types of reflux-valve as sold commercially do not meet this stipulation, § 290. On the contrary, we can be sure that when such a valve is in use and when the liquid column is undergoing rapid retardation after the pump motor has tripped out, reversal of the column will *already have occurred* before the flap-door finally reaches its seat: that is to say, the liquid column is moving backwards at an appreciable speed before it is violently arrested by valve closure. It is easy to form a rough impression of how severe the resulting shock pressure may be. Let us assume (i) that the flap-door of the reflux-valve is so light in weight, or is hung in such a manner, that it responds passively to the movements of the liquid column, (ii) that during normal pump operation the door has an effective opening of 1 in., (iii) that the ratio of the retarding head on the column at the moment of arrest, to the length of the liquid column, $h/l = \frac{AC}{l}$, Fig. 190, has the value $\frac{1}{4}$. From § 262, the resulting mean rate of retardation, $\frac{dv}{dt}$, is $g(h/l)$, viz., 8 ft./sec.² Using the fundamental law of motion, we find that after the column and the flap door have moved 1 in. backwards, from the instant of zero velocity, they will have attained a negative or reverse velocity of 1·15 ft./sec. The resulting water-hammer pressure, as computed from the usual approximate formula for sudden valve closure, has the value 63·4 × 1·15 = 73 lb./sq. in.

Now this pressure is altogether distinct from, and additional to, whatever positive surge was generated by the main pressure-wave in the pipe, Fig. 190. It may therefore be given the distinguishing title *slam-pressure*. The autographic pressure diagrams reproduced in Fig. 192 prove that its value is quite as high as our tentative calculations, above, had led us to expect. The conditions were : 8-in. pump ; large 10-in. foot-valve at bottom of vertical suction-pipe. Diagrams (i), (ii), (iii), total static head = 4 m., length of water column = 11 m. Diagram (iv), static head = 13 m., length of water column = 17 m. It will be noticed : (*a*) that the slam-pressure is very much greater than the normal static head, H_s, on the installation, but that (*b*) as the initial rate of flow through the pump (before pulling out the main switch) is reduced, so also does the severity

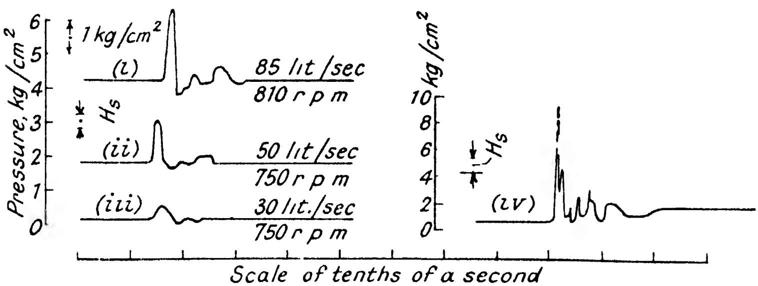


FIG. 192.— Autographic records of slam pressures.

of the slam-pressure fall away, (*c*) that the shock shown in diagram (iv) was so violent that the indicator could not accurately record the peak pressure. These particular diagrams are typical of many pumping systems having relatively short pipes ; the disturbances arising from " slam " are much more troublesome than those that depend on " surge ".

274. Effect of Relief Valve. There is one final way of convincing an observer who might still be sceptical of the punishing effects of uncontrolled reflux-valves. Let him stand by the side of the pumping set during the process of shutting-down represented in Fig. 192 : the hammer-blow is no mere figure of speech—it is quite audible, and the accompanying tremor of the pump and pipework cannot be mistaken. Neither the pump nor the piping can be expected to enjoy the experience.

Paragraphs in Part D of the book, §§ 326-337, describe

various methods of relieving the inertia shocks consequent upon uncontrolled pump stoppage. Here it must suffice to see what the possibilities are likely to be. There are three separate—though not necessarily unrelated—types of surge to be considered : (i) the maximum *negative* surge, (ii) the maximum *positive* surge, Figs. 188, 190, and (iii) the slam-pressure for which the reflux-valve alone is responsible. For a *given* pumping installation working under specified conditions, there is *no* method of controlling the negative surge ; it must inevitably develop. But there are various methods of modifying the installation, or its mode of operation, so as to reduce the intensity of the negative surge.

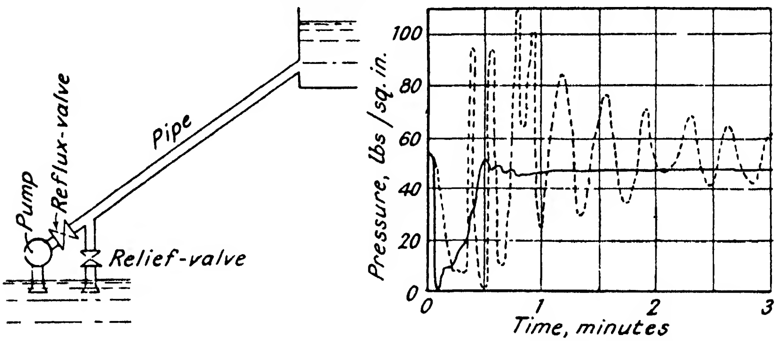


FIG. 193.—Time-pressure diagram, without and with relief-valve, for pipe 12,350 ft. long.

Anything that reduces the negative surge will also reduce the ensuing positive surge. The positive surge may also be very nearly eliminated by opening a relief-valve in the pipe at the same moment that the reflux-valve closes, Fig. 193 ; this allows the positive wave to dissipate itself harmlessly instead of building up an undesirable pressure against the reflux-valve (*). Thereafter the relief-valve is gradually closed in such a way as to bring the reversed liquid column slowly to rest. The general disposition is shown in Fig. 193, which also demonstrates how successful the relief-valve may be in actual use (*).

Before the relief-valve was fitted (broken line), the pressure-fluctuations were serious and persisted for several minutes. After using the relief-valve (full-line), excessive pressures were hardly noticeable.

PART D

INSTALLATION AND OPERATION

CHAPTER XVIII

ALLIED MACHINERY AND AUXILIARY APPLIANCES

275. Partners and Servants of the Pump. The pump is built, tested and ready to work. Presumably we know how it will behave when driven at various speeds, how it will respond to changes of suction lift, to changes of liquid density, and so on. Now there remains the problem of incorporating the machine into a complete pumping installation. That means that other items must be chosen, the whole plant assembled on site, and a maintenance programme framed that should ensure long and successful service. Such is the material that forms the subject-matter of Part D of this book.

In our role of supporters and protectors of the pump we need not be too delicate in specifying which of the additional appliances are partners and which are servants of the pump. In a general way, we must regard everything else in the installation as in some degree ministering to the requirements of the pump and therefore subject to our control and supervision. This complementary equipment may include :—

(i) The motive-unit—the engine, motor, or whatever it may be—that drives the pump or delivers energy to it.

Here we shall certainly be well-advised not to quibble about orders of precedence, but to grant the title of partner without hesitation to the motive-unit.

(ii) The transmission device between motive-unit and pump, e.g. belt, gear-box, coupling, etc., etc.

(iii) Bedplate, foundations, etc.

(iv) Suction and delivery piping, manifolds, etc.

(v) Valves and control organs of all kinds interposed in the piping.

(vi) Auxiliary pumps and apparatus.

(vii) Measuring and recording instruments for use during the operation of the plant.

(viii) Buildings or protective equipment for the installation.

Although whenever possible we shall willingly accept standard material whose design we leave with confidence to specialist manufacturers, yet if the pump's comfort or convenience demands it we may have to frame quite particular specifications.

Motive-units

276. Range of Driving Machinery. Rotodynamic pumps are nearly always driven by one or other of the following machines :—

(i) Electric motors.

(ii) Steam turbines.

(iii) Internal combustion engines.

Reciprocating steam engines are still in use to a small extent, but they are very rarely included in new installations, except on ship-board. Occasionally there may be special conditions in which a water-turbine could drive the pump. When small pumps serve as auxiliaries to larger machines of every kind, they may draw their power directly from the main machine, through shaft, gears, etc.

A primary question to be settled is, what should be the *rated output* of the driving unit ? Shall we flatter the pump by specifying a motive-unit whose B.H.P. output is just equal to the S.H.P. pump input ? Or shall there be a generous margin —20 per cent. or so—to allow for deterioration in the performances of the pump and of the motive-unit ? Chapter XVI has explained some possibilities of decline of pump performance ;

the makers of the motive-unit should be able to tell what to expect from their own products. So the only comment worth adding here is this : if the rating of the motive-unit is a little on the generous side, so that it can be confident of being master of its work no matter what happens—within reason—then nobody need ever hear about it. But the engineer may never hear the last of an undersized unit that is always struggling with adversity. Moreover, as rotodynamic pumps often run continuously night and day, the rating of the driving unit may have to be on a twenty-four hour basis.

The next question is shaft speed and its possibility of variation. Chapter XV has shown how much better the pump behaves if it can choose its own speed to suit changing requirements ; so manifestly a variable-speed motive-unit may be very advantageous. In regard to matching the speed of the motive shaft to that of the pump shaft, a good deal of give-and-take is possible here, with the pump designer frequently in the role of the giver. So long as he is warned beforehand, he can vary the pump speed to suit given conditions either by modifying the shape of the rotor, § 228, or by varying the number of rotors in a multi-stage pump, § 117. When such possibilities are exhausted, the transmission gear, § 284, must bridge the gap in speeds.

The last of these general questions is the direction of rotation of the driving shaft. The installation and operating conditions may require that the pump occasionally runs backwards-way, § 271 ; or at any rate they may indicate that such an emergency may arise and that appropriate provision must be made for it. Will the motive-unit come to any harm if it is forcibly compelled to share this reversed rotation ?

277. Electric Motors. In popular terms, one might say that rotodynamic pumps and electric motors were " made for each other ". Pump rotor speed and electric rotor speed very often—in fact nearly always—permit of direct coupling of the two shafts, and the even torque delivered by the motor is exactly matched by the even torque required by the pump. Only in the matter of speed variation is the electric motor often disinclined to co-operate.

Direct-current Motors. These are naturally the pump's first choice. The range of speed provided by normal regulation

(rheostatic control of shunt field circuit) is adequate for all usual pump requirements.

Alternating-current Motors. (i) *Synchronous.* This term itself has a discouraging sound. How can we expect rotational flexibility when the motor speed is controlled solely by the frequency of the supply-circuit and the number of stator poles? Thus for a 50-cycle circuit the speed range is confined to fixed values such as 1500, 1000, 750 r.p.m., etc.

(ii) *Induction.* Here again standard machines are virtually tied each to a single speed, which is a little lower than the synchronous speed by the amount of the slip. The jump from about 1450 r.p.m. to about 2900 r.p.m. is often exceptionally irksome, for the intervening range may cover just those speeds most likely to be useful for standard single-stage pumps. Also the maximum attainable speed (in Great Britain) of 2950 r.p.m. is sometimes less than the pump maker would like.

278. Variable-speed A.C. Motors. Specially-designed A.C. motors, or standard machines with special equipment, can be provided to give either (i) an infinitely variable range of speed within specified limits or (ii) a choice of two speeds. The types most likely to be useful (*) for driving rotodynamic pumps are :—

(i) *Standard Slip-ring Motor, with Rotor Resistance.* Increasing the external rotor resistance increases the slip and thus brings down the speed. But the whole of the energy absorbed in the resistance is wasted, and moreover, the speed for a given setting of the resistance is influenced by the load.

(ii) *Motor with Pole-changing Device.* The stator windings effectively in use can be changed by a simple switching device, enabling either of two speeds (but no intermediate speeds) to be obtained. Sometimes virtually two separate motors are mounted on one shaft, each being a standard machine, but the number of stator poles being different in the two units.

(iii) *Commutator-type Motor.* Infinitely-variable choice of speeds over a wide range is available, but voltage is limited to about 600.

(iv) *Motor with Frequency-changing Set.* When speed-reduction is desired, the main motor works in con-

junction with an auxiliary motor mounted on the same shaft; this auxiliary motor is energised from the slip-rings of the main motor, through the medium of the rotary frequency-changing set.

(v) *Motor with Scherbius Equipment.** Here also a special rotary unit—a "slip regulator"—is used to obtain speed reduction. Like the frequency-changing set, it gives infinitely-variable speed control.

From these possibilities, the most attractive is the commutator motor, (iii), for the smaller range of powers, and the Scherbius-controlled motor, (v), for large outputs. In any event the final choice can only be made after alternative means of regulating the head and output of the pump have been studied, Chapters XVI, XIX.

279. Special Motors. From the point of view of the motor manufacturer, none of the machines hitherto mentioned rank as special ones: they do not require major adaptations to enable them to suit the pump. But sometimes the electrical motive-unit must be, in whole or in part, non-standard. Some examples are:—

(i) The small motors required for flange-mounted pumps, Fig. 38 (ii).

(ii) Motors for submersible bore-hole pumps, Fig. 92. Of all machines, these depart most widely from standard construction.

(iii) Vertical motors with hollow shafts, adapted for bore-hole pumping plants, § 315.

(iv) Occasionally the motors for large shaft-driven borehole pumps are set at the bottom of a dry well, of which the borehole itself forms the downward continuation. Dissipation of heat is here the problem, and it is solved by providing the entire motor stator with a water-jacket. In other underground pumping-plants, it may be necessary to devise special air-cooling systems for the motors.

(v) Inclined motors for inclined propeller pumps, Figs. 70, 204 (II), may require special study.

280. Switch-gear. Some notes on the starting conditions of rotodynamic pumps have been given in § 266; whatever the type of pump or conditions of installation, the *hydraulic* part

of the torque is invariably *zero* at the actual moment of starting from rest. At this moment, that is to say, the driving motor only needs to deliver to the pump shaft a torque sufficient to move from rest the rotating elements of the pump. It is true that this torque may be more than that required merely to overcome inertia ; if the pump has been standing idle for a long time, the working parts may have become partly rusted up or silted up. When once the resistance of the liquid becomes operative, the rate at which it builds up will depend upon the characteristics of the entire hydraulic circuit ; thus the permissible speed of operation of the motor starting-gear, in order to keep within a stipulated starting current, may be assessed by the methods explained in § 266. Sometimes the favourable starting characteristics of the pumping set permit simpler gear to be used than would otherwise be possible ; for instance, quite large squirrel-cage induction motors may have direct on-line starters.

Electrical operation easily lends itself to such conveniences as (i) push-button starting and stopping, (ii) centralised control of a group of pumps from a single control-post, (iii) remote-control from a distance of several miles. (See also § 295.)

281. Automatic Electric Control. Automatic or semi-automatic working of motor-driven pumps can be secured by the use of elements responsive to changes of pressure, level, or discharge. These elements actuate relays which initiate the train of switching operations designed to start, speed-up, slow-down, or stop the pump motor according to service requirements. In *liquid-level controlled switches* the responsive element may be a float which obeys the movements of the surface level in the reservoir from which the pump draws or into which it delivers. When the float has fallen nearly to the bottom of, e.g. the delivery chamber, the pump motor cuts in ; when the chamber is nearly full, the float causes the motor to cut out. Instead of moving floats, fixed electrodes set at suitable levels can advantageously be used. By these means the surface level can be maintained within whatever range is desired.

Pressure-controlled switches are required when no free liquid surface is available. The basic principle is represented by a Bourdon type pressure-gauge fitted with electrical contacts, and connected to the pump delivery pipe. The pump motor

would start when the pressure fell below a pre-determined limit, the switch-gear being energised by the engagement of the pressure-gauge needle with the "start" contact; similarly when the delivery pressure rose to its upper limit, the needle would touch the "stop" contact.

Flow-controlled switches are essentially modifications of pressure-controlled devices; but they are responsive to changes of *differential* pressure generated in a Venturi meter or the like, interposed in the delivery pipe. In this way the rate of flow of the pump can be kept nearly constant irrespective of fluctuations of total head. Manifestly this must be done not by starting and stopping the pump, but by varying its speed in small steps (*).

282. Steam Turbines. These are nearly as well adapted for driving rotodynamic pumps as electric motors are. Impulse-type machines with a few velocity stages are often preferred, either with horizontal or (occasionally) with vertical shafts. Units direct-coupled to single-stage side-inlet centrifugal pumps, running at speeds up to 10,000 r.p.m., form the most compact of all types of boiler feed pump, § 345. Such turbines are specially built for the purpose. One shaft alone carries both turbine rotor and pump impeller, and it may be supported by two bearings only.

For driving large waterworks pumps and machines of comparable size (*), standard designs of turbine are serviceable, coupled to the pump through enclosed speed-reduction gears.

The question of thermal expansion may arise when verifying the alignment of turbine shaft and pump shaft or gear shaft, § 139, but this time it is the elevation of the turbine shaft that may be affected by temperature changes.

Steam-turbines usually possess the valuable attribute of speed variation.

283. Internal-combustion Engines. Nearly all standard classes of oil engine can find a place in a pumping installation of one kind or another. Horizontal slow-speed units are admirably adapted for direct-coupling to high-capacity, high specific-speed pumps in drainage and irrigation stations. Small vertical single-cylinder engines may drive centrifugal pumps through V-belts, and larger vertical multi-cylinder engines may be direct-coupled to multi-stage pumps installed

in oil pipe-lines. Step-up or step-down gear-boxes can often conveniently be interposed between engine and pump.

Apart from the question of reversed rotation, § 276, perhaps the only item of the driving installation that requires particular scrutiny is the starting equipment. No matter whether the engine is started by compressed air or by a small auxiliary engine, the energy available must be adequate to accelerate the main engine up to the firing speed in spite of the additional inertia of the pump rotating parts. If, because of step-up gearing, the problem becomes intractable on normal lines, then a clutch must be interposed in the transmission system, § 284.

Direct-coupled *petrol* engines are specially suited for portable pumping sets, e g fire pumps, contractors' pumps, etc., § 347.

Very occasionally, a *suction-gas engine* may drive a group of pumps through an electrical transmission system.

Auxiliary Equipment

284. Transmission Systems. In choosing any apparatus for transmitting power from motive unit to pump, we can always claim at least that the pump is "easy on the drive"; its perfectly uniform torque characteristics ensure that.

Direct-coupled Sets. When the shafts of pump and of motive unit are coupled co-axially together, as so often happens in electric pumping sets, a so-called elastic or flexible coupling is nearly always essential. Rigidly-bolted flanged couplings are usually unsuitable not only because of the side loads on the shafts that may arise through imperfect alignment, but because the shafts may like to have freedom to adjust themselves individually in an axial direction. Thus the automatic adjustment of a multi-stage pump with balancing disc, § 124, would be ineffective if longitudinal constraint were put upon the shaft. The standard pin and rubber bush type of coupling serves very well for low and medium powers, and the steel-spring type for heavier duties.

Shaft Drive. The most familiar example of a long *vertical* driving-shaft interposed between motive unit and co-axial pump occurs in shaft-driven borehole pumps, § 132. *Horizontal* units may be coupled by a short shaft when the set is handling volatile and inflammable liquids which require the pump and pipes to be

isolated in a gas-proof chamber. The shaft bringing power from the external motive unit is protected by a packed gland at the point where it passes through the partition wall.

Hydraulic Couplings. These also are used for connecting two co-axial shafts, but the pump shaft and motive-unit shaft no longer revolve exactly at the same speed. The slip, or percentage reduction of pump speed in relation to the speed of the driving shaft, may be controlled by regulating the amount of oil in the coupling. If the coupling is full, then the slip may only be about 2 per cent.; and the maximum slip with the coupling partly drained rarely exceeds 20 per cent. Torque is transmitted without change; but since power is the product of torque and speed, it is clear that the pump power input is invariably less than the motive-unit power output. The energy difference is wasted in heat, and consequently means must be found for dissipating the heat.

This variable-speed device is only likely to be used when constant-speed A.C. motors are required to drive variable-output pumps. The energy then lost in the coupling at low pump outputs can be shown to be much less than the energy that would be wasted in a throttle valve if a constant-speed direct-coupled pump were used, § 240. **(Example 36)**

Friction-clutches are used chiefly to overcome starting difficulties when high-speed oil engines drive multi-stage high-pressure pumps through step-up gear-boxes. Hydro-mechanical couplings have been installed for making and breaking the connection between very large electric motors and pumps in hydraulic storage installations, § 350.

285. Indirect Transmission Devices. This term embraces the systems that may be needed when the pump speed is different from the speed of the motive unit. No longer is it possible to keep the two shafts in line: on the contrary, they may be spaced some distance apart.

Flat Belt. This type of drive (leather, camel-hair, etc.) has given excellent service in the past, and it will probably be a long time before it is wholly superseded. It is particularly advantageous in rough conditions when it is not possible to ensure accurate alignment of the two shafts. A good example is offered by an irrigation pumping plant where a slow-speed horizontal oil engine running at a few hundred revolutions per

minute drives a pump running at 1000 r.p.m. or so. The effect of the heavy belt pull on the pump bearings must not be overlooked. One, or possibly two, extra outboard bearings may be desirable, e.g. Fig. 39 (iii), § 71.

Multiple V-belt. This neat and compact type of drive is rapidly gaining popularity. By its use, very attractive little pumping sets can be built up from a vertical oil engine and a pump, mounted on a common bedplate.

Gear-box. These units have a very wide range of application. As the gears are totally enclosed and run in oil, transmission efficiencies up to 97 per cent. are feasible. Step-up (speed-increasing) and step-down (speed-reducing) types are available, with input and output shafts either parallel or at right-angles one to the other. Speed-increasing gear-boxes with shafts at right-angles may transmit power from oil engines to shaft driven borehole pumps. As it is just as important to shield the gear shafts against axial loads as it is the pump and motive-unit shafts, there must be a flexible coupling on both the input and the output side of the gear-box. Perhaps the only type of gear-box that falls outside the manufacturers' normal range is the inclined one shown in Fig. 204 (II).

286. Bedplate, Foundations, etc. For stationary pumps of small and medium size, a simple combined bedplate for pump and motive unit is all that is required, Fig. 38 (i), the complete set being bolted direct on to a concrete foundation block. The bedplate may be of cast-iron or welded steel. When the pump has a downward-pointing suction branch, Fig. 39 (i), or underhung transfer pipes, Figs. 79 (i), 81, then manifestly the shape of the bedplate will be rather more elaborate. For larger units there should be special foundation pads to receive brackets cast on the pump body, Figs. 39 (iv), 66; or the brackets may rest on longitudinal steel or cast-iron girders which are themselves supported by the foundation pads.

In regard to the size of the concrete foundation block, it has to be remembered that although motor-driven pumping sets nominally run without vibration, there is a chance that in course of time vibration may set in as a result of wear or corrosion of the rotating element, § 259. In a works or industrial establishment, such vibration would be regarded as harmless, whereas it would be objectionable in a business or residential

building. In this event a specially heavy foundation block might be desirable, or else some means of insulating the block from the building itself. If the motive unit, e.g. an internal-combustion engine, itself incorporates unbalanced elements, it is manifestly essential to observe all the precautions such machines demand. On the other hand, pumping sets intended for rough outdoor use can often do without foundations altogether; such are multi-cylinder engines driving multi-stage pumps, mounted on heavy longitudinal steel skids.

287. Suction, Delivery, and other Piping. There are at least three different aspects from which the piping system may usefully be studied: (*a*) The effect on pump performance has already been reviewed in §§ 240 to 244, (*b*) the pipe size may have an important bearing on the economics of the installation, (*c*) when large, low-head installations are in question, the pipes and pipe-bends are so bulky in relation to the size of the pump that the run of the pipe-work may dictate the type of pump to be chosen.

(i) *Pipe Size.* Aspect (*b*), above, comes into prominence when pipe friction constitutes a large proportion of the total effective head on the pump. For a stipulated discharge, a generous pipe diameter will mean a low pipe velocity, a low friction loss, a relatively low total head, and therefore minimum pumping costs. But on the other hand the capital charges debited against the pipe will be considerable. An inexpensive pipe of restricted diameter would raise pumping costs unduly; thus the optimum pipe diameter that ensures minimum *overall* cost can only be found after a series of computations involving a good deal of trial and error.

(ii) *Suction Piping.* The suction pipe is often made of larger diameter than the delivery pipe so as to keep down inlet losses and thus to increase the permissible suction lift, § 253. For the same reason, the suction pipe should be as short and straight as possible; it should rise continuously all the way to the pump, so as to give no lodgment to bubbles or pockets of air. If this is found to be impracticable, then residual air must be allowed to collect in an air vessel from which it is periodically evacuated, Fig. 199. The intake to the suction pipe (if unprotected by a foot-valve) should be rounded or bell-mouthed.

Because the suction pipe has *nominally* to withstand only

a few feet of suction head, it does not thereby follow that it can be of lighter construction than the delivery pipe. If a foot-valve is fitted, as in Fig. 188, the *positive* pressure in the suction pipe near the foot-valve may be greater than anywhere else in the system. Immediately after the valve has closed, this cumulative pressure may momentarily include the full static pressure-head, augmented by the surge pressure, Figs. 188 or 190, and by the slam-pressure, Fig. 192.

288. Piping Systems (*continued*). (i) *Junction-pieces, manifolds, etc.* It will only be by chance that the most favourable diameter of piping will exactly match the diameter of the pump branches. In designing the conical or diverging junction-piece that couples the pump to the *delivery* pipe, it is well to remember that this component can itself be regarded as an extension of the conical recuperator, Fig. 28 (*b*), § 45 (*b*). It therefore gives the possibility of quite an appreciable regain of pressure-head. As for the corresponding taper reduction-piece on the *suction* side of the pump, it may have to be of asymmetrical form so as to give no opportunity for accumulations of air.

When a number of pumps work in parallel, careful shaping of the inlet and outlet *manifolds* will help to minimise energy losses due to eddying and turbulence, Figs 209, 210.

(ii) *Supporting the Pipes.* The suction and delivery pipes should preferably be so supported that they cannot transmit to the pump flanges excessive thrusts. Thus, in a faultily-disposed pipe system carrying hot liquids, thermal expansion and contraction of the pipes might actually distort the pump casing and throw the whole set out of alignment. Overhung arrangements, e g. the pumps in Figs. 38 (i) and 39 (i), should be specially watched. In buildings where the piping cannot be allowed to transmit any perceptible sound or vibration outside the pump chamber itself, special sleeves of reinforced rubber or similar sound-absorbing material may be interposed between the pump branches and the pipes.

(iii) *Auxiliary Piping.* To permit the pumping set to keep a neat and tidy appearance, small-bore pipes will be needed to drain away leakage liquid from glands, vent-cocks, etc. There will be a continuous leakage flow to be disposed of, if a multi-stage pump has a hydraulic balancing device, § 124 ; while

if there are in addition water-cooled bearings and the like, the auxiliary piping system will grow fairly complex. Quite apart from this array, one might find water pipes for water-sealed glands, lubricating pipes conveying oil or grease to the main bearings, and air pipes for the priming system, § 292.

289. Valves. In a normal pumping installation, there may be

(i) Isolating valves.
(ii) Regulating valves.
(iii) Reflux or non-return valves.
(iv) Small auxiliary valves or cocks.

Standard sluice-valves or full-way valves usually serve as *isolating* valves, although for large low-head installations butterfly-valves may be preferable. Special requirements can be met by plug-valves, § 337. Hydraulic or electric operation is essential for important plants. Automatic power-operated gate-valves are described in § 330. All such valves are intended to be kept fully-shut when the pump is stopped, and fully-open when the pump is running normally. They should preferably be interposed both on the suction and the delivery side of the pump, so that they can isolate the pump from the rest of the system and permit the unit to be opened for inspection or repair.

Regulating-valves control the discharge of the pump by throttling, § 240, and therefore they inevitably waste energy. In principle, then, only relatively small systems should be so regulated. Standard sluice-valves usually serve, set on the delivery side of the pump. Needle or streamlined valves may be preferable for isolating the pump or for regulating the flow. If a pump handling aggressive water is run for a long time against a partly-closed sluice-valve, the resulting cavitation in the valve itself may ultimately destroy it.

Auxiliary Valves, Cocks, etc. As these usually form part of the priming system of the installation, they are enumerated under that heading.

A drain-plug at the lowest point of the pump casing is useful when the time comes to open up the pump for inspection.

290. Reflux or Non-return Valves. These automatically prevent return flow of liquid when the pump is stopped; they all embody a gate, flap, or disc which swings open when the liquid flows normally through the system, and which shuts

immediately there is any tendency to backwards motion. To some extent all such accessories must be regarded as necessary evils. At the best they exact a continuous toll of energy from the liquid that traverses them; at the worst they may set up "slam" pressures high enough to damage the system, § 273. The following notes relate to more or less standard types of reflux valve: special designs adapted to unusually difficult conditions are described in § 333.

(i) *Foot-valve.* This is the name given to the non-return valve when it is fixed at the bottom of the suction pipe; it is

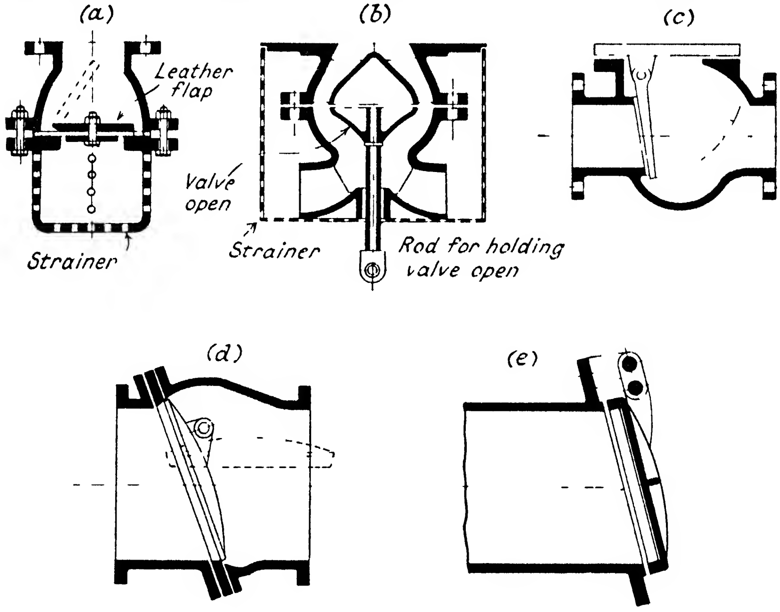


Fig. 194.—Types of reflux valve.

preferred only for comparatively small installations. Of the two examples illustrated in Fig. 194, type (*a*) is the commonly-used one. Sometimes the door has a leather seating, sometimes it shuts metal-to-metal. The complete assembly usually includes a *strainer*, either of cast iron or perforated steel, for arresting floating material that might otherwise enter and clog the pump. The head loss occasioned by the foot-valve and strainer may amount to two or four times the equivalent velocity head in the suction pipe; it may be much greater if the strainer becomes choked.

The improved type (*b*) in Fig. 194 has two advantages: the waterway is of streamlined form, and the resistance to flow can be made still lower by mechanically holding open the valve. Unless the holding-up device can be automatically tripped before the pump is stopped, e.g. as in § 333, there is, of course, the danger that the attendant may forget to release the valve and so permit return flow.

(ii) *Reflux Valve.* Although this is a general term for all types of non-return valve, it is more especially used when the valve is set on the delivery side of the pump. In this position it is better able to protect the pump and suction system against inertia surges in the delivery pipe than a foot-valve could possibly do. Diagrams (*c*) and (*d*) in Fig. 194 show respectively a reflux valve with a normal type of hinged door, and one with a tilting disc which has certain advantages.

(iii) *Outlet Flap-door.* Used only in low-head installations, the outlet flap-door is fitted to the end of the delivery pipe, Fig. 194 (*e*).

291. Priming Equipment. A pump installed in the customary position above the suction well has not itself the power of initiating flow of liquid *so long as the pump remains dry.* The reason was made clear in § 236: when the rotor has only air to work upon, the differential-pressure generated is negligible in relation to the designed working pressure of the pump. No matter for how long the pump were to be run, the water would never rise more than an inch or two up the suction pipe. It is for overcoming this difficulty that *priming* apparatus is essential; its purpose is to fill the pump casing and the suction system with liquid before the shaft is rotated at all. In any event the basic problem is to get the air out and to put the liquid in. There are two ways of doing this: (i) we can pour liquid into the pump casing from above, and allow the displaced air to escape, or (ii) we can exhaust the air from the casing, which will cause the liquid to rise up the suction pipe and eventually fill the casing.

System (i) is manifestly only admissible when the suction pipe has a foot valve, § 290 (i), for otherwise the liquid would run out of the pipe as fast as it was poured in. A funnel with cock admits the liquid to the casing, and an air vent at the highest point allows the displaced air to escape. Makers do

not remember as often as they might that two separate passages are needed ; it is not reasonable to hope that one small opening will serve at the same time for incoming water and for outgoing air. The correct disposition, which is suited for small installations only, is illustrated in Fig. 198.

292. Evacuating Appliances. Medium- and large-sized pumps must necessarily be primed by the exhausting system (ii) above. The choice of evacuating apparatus usually lies between *ejectors* and *pumps*. If steam, compressed air, or even exhaust gases from an internal-combustion engine are available, an *ejector* can be used of the type shown in Fig. 195. Diagram (i) shows the ejector, with its isolating-valve, mounted above the highest point of the pump casing. A water-gauge tells the attendant when evacuation is complete. The main valve in the pump delivery system is, of course, shut during the whole priming operation. Diagram (ii) shows the internal construction of the ejector itself.

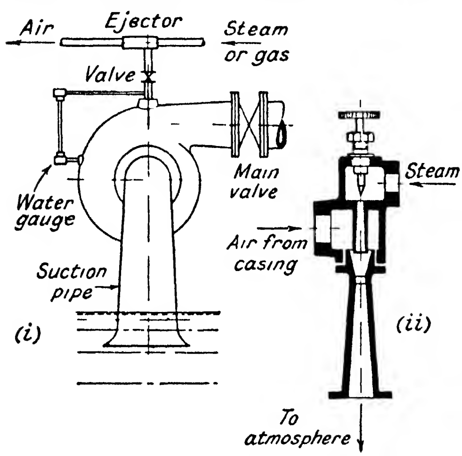


FIG. 195.—Priming ejector and its use.

Suitable types of *evacuating pump* include :-

(i) Reciprocating piston.
(ii) Rotary sliding-vane.
(iii) Liquid-ring.

Reciprocating pumps are virtually standard types of dry vacuum pumps, having one, two, or even four cylinders. As they only work intermittently, and never have to draw against a high vacuum, refinements of construction are unnecessary.

Rotary sliding-vane exhausters are likewise more or less standard products.

Liquid-ring pumps were described in § 153 (*b*).

The *capacity* of the evacuating set should be such that any

one main pump can be primed and made ready to run in about five or ten minutes.

The method of *drive* will depend upon the motive system used for the main pumps. If electrical energy is available, the auxiliary pumps will certainly be driven by small electric motors, either direct-coupled or through belt or similar transmission.

Reciprocating pumps must be shielded against the risk of *flooding*. If, after the main pump casing is evacuated, it is still in free communication with the exhausting pump, water will certainly enter the cylinders and smash the pump—for we are assuming that it is still running. The customary safeguard is a float-valve interposed in the air line; when water rises into it, a ball-valve floats upwards and shuts off access to the vacuum pump. Greater security is given by the loop in the primary pipe seen in Fig. 199. Sliding-vane pumps may require similar protection, but small units can sometimes admit water without damage. Liquid-ring pumps, of course, can work with water as easily as with air, § 153 (*b*).

There is a great diversity of ways of connecting the exhausting pump to the main pump, in addition to the independent electric drive mentioned above. Close-coupled systems come under the heading of self-priming pumps, § 156; other systems are mentioned in §§ 339, 340.

293. Other Auxiliary Appliances. (i) *Drainage Pumps.* In large, low head installations, auxiliary pumps are desirable for drying out the inlet and outlet conduits, etc., when a main pump is laid out of service, so that the passages may be cleaned out and the pump examined, overhauled or repaired. Vertical motor-driven units, drawing from a sump at the lowest point of the system, are often very convenient for this duty.

(i) *Cooling Pumps.* Main pumps having water-cooled bearings, stuffing-boxes, bedplates, etc., §§ 139, 143 may require auxiliary pumps for handling the coolant. A gear-wheel pump or similar positive rotary unit would probably be suitable. The coolant need not be run to waste, but could continuously be circulated through a heat-exchanger.

(iii) *Lubricating pumps* are sometimes needed. They may force oil to the external bearings of centrifugal pumps or grease

to the internal main bearings of half-axial or axial type pumping units.

(iv) An *auxiliary generating set* will be necessary if important main pumps are driven by oil engines or by steam turbines; electric power must be provided for lighting and for the engine and pump auxiliaries.

(v) If dirty water flows through the main pumps, *filters* may be required for furnishing clear water to the water-sealed gland, § 86, or to the sealing rings, § 151.

294. Indicating and Recording Instruments. The instruments required to show the pump's behaviour on the test-bed were enumerated in §§ 166 to 170. Which of these should be chosen for checking the pump's performance after it has been installed? The choice must be guided by one important limitation: whereas the observations in the works could be interpreted by an experienced testing staff, the indications on the site must be made unmistakably clear to an unskilled attendant—or they must indeed be automatically recorded. Keeping this and other fairly obvious points in mind, we can now run down the list of apparatus already compiled, Chapter XII, and prepare a new list suited to installed conditions.

Speed Belt-driven or direct-coupled tachometer mounted on the pump, or electrical tachometer with panel-mounted dial.

Head or Pressure. For low heads, open water-columns with floats actuating pointers, or with floats electrically connected to remote-reading dials.

For borehole pumps, " bubble " type pneumatic gauges, or special electrical depth-recorders.

For general use, standard types of Bourdon, diaphragm, or bellows dial gauges, graduated in units of head, pressure, vacuum, etc.

Discharge. The Venturi meter is nearly always preferred for important installations, but the " Helical " type of inferential meter is often worth considering. Devices specially suited to rotodynamic pumps that require no additional apparatus interposed in the pipe-line itself, are: (i) the pipe-bend meter, and (ii) the Annis suction-pipe meter, § 179.

Power Input. As a rule it is only possible to gauge the input to the pump by interpreting the motor instrument readings of electrically-driven sets.

Types of instrument not mentioned in Chapter XII include :—

Water-gauge (standard type) mounted on upper part of pump casing. This very useful accessory shows when all the air has been exhausted during the priming operation, § 292.

Thermometers are sometimes desirable when cooling systems for bearings, etc., are involved.

Gate-opening indicators in large and complicated installations show at a central control station the setting (open or closed) of each sluice-valve or sluice-gate.

295. Some Typical Groupings. Although the indicating apparatus on the site is primarily intended to show what the pump is doing at any given moment, yet we may be equally interested in observing how the performance changes in course of time. If, because the necessary instruments are lacking, the pump is allowed to fall into bad condition, the resultant continuous waste of energy may represent a much greater sum than was saved on the instruments. Moreover, minimum operating costs can only be attained by trying to keep the pump performance more or less delicately poised upon a " ridge of maximum efficiency ", § 222 ; and this in turn is only possible if the operating staff are thoroughly informed about each aspect of the pump's behaviour. It follows that since it is in the high power installations that the biggest opportunities of economy occur, it is there that the most complete instrumentation can be expected.

As soon as the cost of the pumping-set is such as to justify any instruments whatever, the first accessory to think about is a *pressure-gauge.* If this is intelligently watched, it not only shows what pressure the pump is generating but it may give quite a useful impression of the rate of flow. At any rate it ought to tell the attendant whether the pump is delivering anything at all. He may be very glad to know this, because small pumps especially may " lose their water " (as a result of air leaks or drop of suction level) without any manifest sign. Next in importance comes an *ammeter*, if the set is electrically driven ; for the ammeter reading and the pressure-gauge reading taken in conjunction may permit the discharge to be still more closely estimated. Still larger pumping sets might

each have, in addition, a *flow-meter* for the group, and *tachometers* for the individual variable-speed pumps.

Developments in the electrical transmission of signals have made possible the grouping of indicating apparatus at a central point, conveniently within sight of the attendant who has under his hands the control elements of the electrically driven pumps themselves, § 280. A single panel may have dials showing respectively speed; pressure, head or level; and rate of flow; and associated with the more important indicators there will be automatic recorders which provide a permanent record on a daily or weekly chart. In a pumping station of major importance there may be a range of such panels (*) which provide additional information such as the setting of the main valves, § 289.

296. Buildings, Pump-rooms, etc. Waterworks engineering has established for itself a tradition of housing its mechanical equipment in substantial buildings that possess—or are believed by their proprietors to possess—some degree of architectural merit. One cannot be sure that this praiseworthy tendency is being maintained in respect of other kinds of pump-house, e.g. those containing drainage or irrigation pumps. Neither the main pumping units themselves nor their accessories are in such installations of a shape that need inevitably impose ungainly peculiarities on the protective building; there is no functional reason why the building should not be seemly and spacious. So it is to be hoped that the engineers and architects concerned will not be deterred from erecting buildings that worthily express the intimate relation between the installation and the countryside it is designed to serve.

Only a few technical points can be mentioned here.

(i) In large, low-head stations the inlet and outlet conduits are so big that they must be formed beneath the pump-house floor. In extreme instances the pumps themselves are so situated. Such dispositions naturally have a direct bearing on the lay-out of the foundations for the building.

The intake conduits must often be protected by screens or grids to exclude floating material that might otherwise choke the pump rotors. Power-operated self-cleaning screens of the travelling-band type may sometimes be recommended.

(ii) The capacity of the travelling-crane, and the head-room

below it, may require special study when the building houses borehole pumps. In erecting and dismantling such pumps, the total load to be carried by the crane may be relatively high, and it must be lifted through a substantial distance. An electric crane capable of rapid and accurately controlled movements will certainly be desirable, § 313.

(iii) Pumps handling volatile and inflammable liquids should be set in chambers isolated from the rest of the equipment by flame-proof partitions; power is transmitted to each of the pumps by a horizontal driving-shaft passing through a packed gland in the partition wall, § 284.

(iv) When large pumping units working in tropical climates are installed in chambers below ground-level, natural ventilation may be inadequate. To maintain tolerable temperatures, large-capacity fans should continuously draw in fresh air, and distribute it through a properly-contrived trunking system.

CHAPTER XIX

THE COMPLETE PUMPING-PLANT

297. A Final Survey. The basic items of information required when choosing or designing the pump itself were summarised in § 63. Various supplementary types of pre-requisite data were mentioned in § 65. In the light of the study of pump performance that has been carried through Part C of this book, the list of controlling factors (*) may now demand amplification or completion in some such manner as this :—

(i) What is the *range* of discharge ? The maximum and minimum, as well as the mean value, should be stated.

(ii) The range of head should likewise be given, viz., minimum, mean, and maximum values.

(iii) The suction conditions must be precisely defined : ruling and maximum values of suction lift, etc.

(iv) What type of service will the pumping-plant be required to give ? Will the pump work (*a*) continuously at or near its design point, (*b*) continuously, but at varying rates of discharge, or (*c*) intermittently, with prolonged periods of idleness ?

(v) With what kind of piping system will the pump be linked ?

(vi) Must a single pump take all the load, or can the duty be shared between a group of pumps ?

(vii) What type of motive-unit will drive the pump ?

(viii) Where will the pump be situated ? In a town or city,

or in a remote country area? What will be its altitude above sea-level?

(ix) Will skilled attendants be available to look after the pumping-plant and to carry out repairs? Or must these duties be left to local unskilled labour?

298. Overall Cost of Pumping. A variety of possible schemes can often be framed to meet the comprehensive conditions just enumerated (*). If there is no other over-riding consideration, the scheme ultimately adopted should naturally be the one that ensures the lowest attainable annual overall cost of pumping. This total annual expenditure can be analysed very crudely into two main items, (i) capital charges, (ii) pumping charges. Item (i) will be directly related to the first cost of the installation; item (ii) to the fuel consumption or power consumption.

The *cost of the pump* is generally subject to the following trends: considering only machines of constant *shape number* and constant *speed*, the pump cost $£_c$ depends roughly upon the *square* of the rotor diameter D, viz., $£_c = K_cD^2$. Now the head generated by such pumps varies as the square of the diameter, and the discharge as the cube of the diameter; thus the power input depends upon D^5. If T represents the number of hours per year during which the pump runs, then the annual *cost of energy* can be written :—

$$£_p = K_pTD^5.$$

Adding together the two terms indicative of items (i) and (ii) above, we find that :—

$$\textit{Annual overall cost} = K_cD^2 + K_pTD^5 \quad . \quad (19\text{-}1)$$

The significance of this expression is clear. Whatever the numerical value of the constants K_c and K_p may happen to be for a particular class of pump, it remains true that the ratio $\frac{\text{energy cost}}{\text{capital cost}}$ increases rapidly as the pump becomes bigger. This suggests that for small plants we should concentrate on keeping down the first cost of the pump; a relatively simple and inexpensive machine will give the most satisfactory service. In large and important installations, on the other hand, in which the pumps are kept constantly at work, it is inevitable

that the power costs will be heavy. A high and a consistently-maintained pump efficiency must therefore be sought by all reasonable means. A refinement of construction that buys a gain in efficiency of 1 per cent. may be quite a profitable investment. The more rigorously the pump performance is controlled by recording instruments, § 295, the more quickly can deterioration be detected and arrested, § 259.

In comparing alternative projects for such installations as these, we should not forget that the cost of the pump may not by any means be the major item in the total cost of the plant. A final decision will only be possible when the two main groups of expenditure incurred *in the whole installation*—capital costs and running costs—have been broken down into individual items and each item separately estimated.

299. Choice of Motive Unit. Because of the overmastering need for minimum power costs in important plants, detailed consideration is essential before the most suitable type of motive unit can be fixed upon (*). Here are comparative comments additional to those offered in §§ 276 to 283 :—

Electric Motors. In bargaining with the electric supply undertaking for a bulk supply of energy, the location of the pumping station will be a matter of some importance, § 297 (viii). If it entails the running of a special new power line over a distance of several miles, the power cost will manifestly go up. But the additional cost may be justified if the station can thereby be run unattended, operated wholly by remote control. On the other hand, the power company could hardly offer attractive terms if a low load factor were contemplated, i.e. if the pumps were only wanted for a few weeks annually. Concessions on the part of the pump user that might be worth his while include : (*a*) to run the pumps as far as possible only during off-peak hours, (*b*) to use synchronous instead of asynchronous A.C. motors.

A particular problem of electric drive occurs when a range of pumping plants is scattered over a wide extent of country remote from existing power supplies, § 322. If all the pumps are controlled by a single authority, then it may be profitable to build one or more central generating plants, thermal or hydro-electric, and to transmit energy to the individual electric pumping stations by power lines laid out for that specific purpose.

Irrigation or land-drainage problems can sometimes be successfully solved in this way.

Oil Engines. In populated areas where there may be direct competition between oil and electricity, the decision may often depend upon a sagacious forecast of future trends in fuel prices. An installation of oil-engine-driven pumps is immune from those risks of total failure of energy that cannot be eliminated in plants fed from public supply electricity undertakings. But its maintenance charges will inevitably be higher than in the equivalent motor-driven station. Oil engines have no rival when small portable plants are concerned, or when isolated pumps of medium power have to be driven (*).

Steam Turbines. If raw coal is to be the source of energy, and if the pump output is considerable, there is no alternative to a steam-raising plant supplying steam-turbine-driven pumps. Such installations may enjoy one advantage over a comparable electric generating plant. They require no circulating pump or special circulating-water system. Provided the head is not excessive, the water from the main pumps can be diverted through the condensers.

300. Comparative Schemes. After the type of motive unit has finally been established, there still remain competing dispositions of plant to be reviewed. A few examples are :—

(i) *Horizontal or Vertical Shaft.* The compactness of the vertical disposition, § 72, is reflected also in the building that accommodates the pumping sets. A relatively narrow pump-house in turn implies a lower price for the travelling crane, § 296. If a side-inlet centrifugal pump of medium specific speed is chosen, then not only will the highest possible pump efficiency be attainable, but the whole piping lay-out will be direct and simple, §§ 311, 336.

(ii) *Single-stage or Multi-stage Pump.* For a medium-power installation generating a medium head, using a specified type of oil engine, alternative possibilities are : (*a*) Engine direct-coupled to a multi-stage pump, (*b*) Engine driving a single-stage pump through step-up (speed-increasing) gear-box, § 285.

(iii) *Direct Coupling or Hydraulic Coupling.* Because of its inexpensive construction, let us suppose that a constant-speed electric motor has been chosen. Variations in the pump head or discharge can be obtained by (*a*) throttling, §§ 240, 241, or

by (*b*) interposing a variable-slip hydraulic coupling between the motor and the pump, § 284. Will the cost of the coupling be worth the energy it saves? **(Example 36)**

(iv) *Variable-speed A.C. Motor or Two-speed Gear-box.* The range of water-levels or of head is here assumed to be so great that some means of regulating the pump speed is deemed to be unavoidable, § 242. Although a variable-speed or a two-speed A.C. motor might be direct-coupled to the pump, § 278, there remains another alternative: if in any event a gear-box between motor and pump is essential, it could be arranged to give either of two reduction ratios, thus enabling a constant-speed motor to offer the pump a choice of two speeds.

(v) *Variable-pitch Propeller Pump or Multiple Pumps.* The comparisons in § 244 took no account of capital cost or of operating convenience. Reviewing again the available possibilities if constant-speed pumps have to force a variable discharge against a constant head, we find: (*a*) For the given duty, an installation comprising a single large pump will always be less expensive and slightly more efficient than an installation of multiple smaller pumps of identical design, §§ 227, 298. (*b*) But will that advantage be maintained if the large pump is burdened by the additional gear required to vary the blade pitch while the pump is running, § 104? (*c*) In any event the station must contain one or more stand-by units, and if all the units are to be alike the principle of a single big pump then becomes much less attractive. It would involve an *installed* capacity double the normal capacity.

For other competitive proposals, see §§ 318-321.

301. Manufacturing Considerations. If in the end the pump selected is below a certain size, the question of cost may still control its actual shape and form; the tendencies exposed in § 298 will probably show that a specially-built machine would be needlessly expensive, and that the nearest stock machine the maker can offer must be accepted. The degree of adaptability provided by the manufacturer's range of patterns, § 65 (iii), can be attained in various ways. Standard frames and pump casings can be disposed to suit the client's system of pipe-work by the arrangements explained in §§ 70, 72. A particular complete pump can be offered for quite a considerable range of head and discharge, *so long as* the basic

relationships of § 218 are enforced. To enable a given pump casing to serve for different duties, it may be fitted with one or other of a series of impellers, having either (*a*) different widths, or (*b*) different diameters, or (*c*) different blade angles. The range of variation would be just such as would not unduly affect the gross pump efficiency.

It will not matter to the client if the machine offered him does not work exactly at the design point of its characteristic. The resulting slight decline in the efficiency of a small pump could be tolerated for the reasons given in § 298.

302. Selection Charts. Modified types of characteristic curve assist the maker in the routine work of choosing a suitable

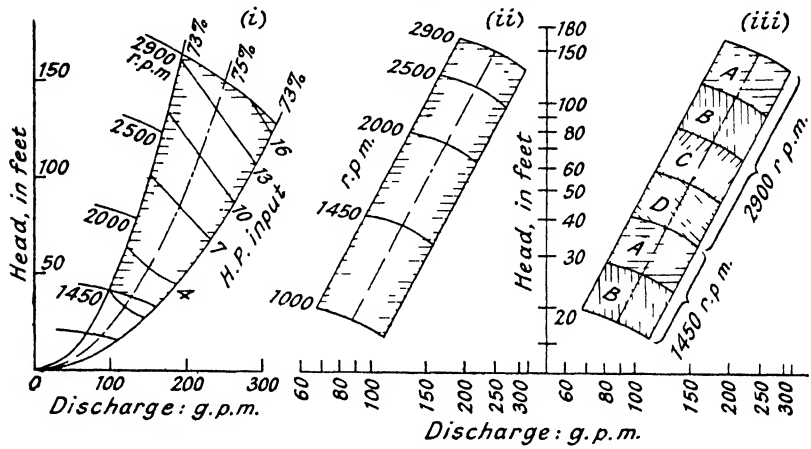


FIG. 196.- Types of linear and logarithmic selection chart.

pump for his customer. Suppose the problem is to establish the range of duties for which a given standard pump, unaltered in any detail, will serve. If *speed-variation* is allowable, the normal set of head-discharge curves might first be plotted, as in Fig. 196 (i), together with a typical discharge-efficiency curve, § 221. This would permit the evaluation of the minimum and maximum percentage flows that would maintain the efficiency within (say) 2 per cent. of the maximum efficiency at the design point. Drawing from the two limiting points the two parabolic iso-efficiency curves shown in diagram (i) we thus delimit the effective area of the chart distinguished by hatching. If desired, additional lines of equal power input can be plotted from basic curves such as Fig. 149. On learning from the client what

head and discharge he wants, the manufacturer can at once note from the chart whether the corresponding point falls within the effective field of the particular model represented ; if it does then the necessary speed and power input can immediately be read off.

Logarithmic division of the co-ordinate axes makes such charts still more convenient, for now all the parabolas are represented by straight lines having a uniform inclination, Fig. 196 (ii).

By far the majority of the smaller-sized pumps now in question will be driven by *constant-speed* A.C. motors. How, then, can such a field as Fig. 196 (i) displays be covered if speed variation is inadmissible ? Instead of expecting a single standard model to do the work, the maker must now provide a range of models, and in each of the models provision must be made for slight variations of impeller diameter. Let the original pump linked with Fig. 196 (i) be termed model *A*. Then its range of duties at standard speeds of (say) 2900 and 1450 r.p.m. would be represented by the appropriate lines in Fig. 196 (iii). Each line could be broadened into a belt by turning down the pump impeller as described in § 181 ; this reduction in diameter, if not carried too far, has virtually the same effect as lowering the shaft speed. The process must stop when the pump efficiency begins to be perceptibly affected. The limits are shown on the diagram, which now has two small fields allocated to pump *A*, in place of the original large one. A new model of pump, denominated *B*, must cover the areas *B* of Fig. 196 (iii), while still a third, *C*, and possibly a fourth, will be needed for the remaining areas.

303. Composite Selection Charts. The process depicted in Fig. 196 has been described in general terms only ; but it should now be easy to see how the manufacturer's range of types can be still further extended. In Fig. 197 there are shown the fields of the pumps *A*, *B*, *C* and *D*, together with particulars of pumps for bigger discharges, *E*, *F*, etc. At the other side of the chart the performance of multi-stage pumps is plotted ; for a given small discharge, variations in head can here be realised by varying the number of stages. As for the general character of the pumps, it will be understood that in moving across the chart from left to right we encounter pumps of higher

and higher specific speed. In this context the term specific speed is advisedly preferred to the term shape number. The latter expression, being descriptive of the *shape* of the rotor, should be independent of random variations of head or discharge, whereas the whole point of the selection charts is to show how a *given* rotor can be capable of operating over quite an appreciable range of specific speeds, § 202.

It is instructive to study the kind of modification which permits the four types of pump, A, B, C, and D to cover the field allotted to them in Fig. 196 (iii). The four rotors will have neither the same shape number nor the same specific speed. On the contrary, as the head goes down the shape number will progressively rise. In order to generate the same form of characteristic, with the shaft turning at an unvarying speed (in this instance 2900 r.p.m.), it is probable that the blade angle will likewise remain unchanged, the diameter will diminish, and the width ratio will increase.

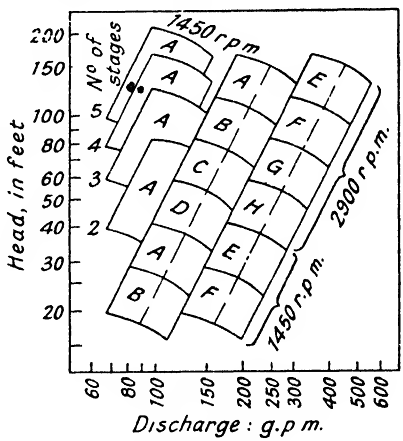


Fig. 197. - Composite selection chart.

For showing the capabilities of the entire range of standard pumps that a maker must nowadays be expected to provide—if necessary at a few hours notice—highly elaborate versions of the selection chart plotted in Fig. 197 may be necessary. Probably it will be expedient to prepare a number of charts, each one relating to a particular speed.

Some Typical Installations

304. Classifications. Up to this point, Part D of this book has steadily exposed the multiplicity and the variety of the allied and auxiliary units that may be embodied in the pumping plant. The whole of Part B of the book showed equally clearly that numerous types of pumps were available. Evidently, then, the total number of possible combinations of

equipment must be very great indeed. In selecting a few representative ones for illustration in this chapter, all that can be attempted is to consider only those that are really entitled to the name " pumping installation ". The pump is the central feature of the plant ; and the motive unit, the building and the auxiliary appliances are specifically arranged accordingly. Such plants are those intended for water-supply undertakings, land-drainage, irrigation, dock-pumping, and the like.

(Examples 37, 39)

In the next chapter (Chapter XX), some particular arrangements of components to meet various particular conditions are discussed, and specific examples are illustrated.

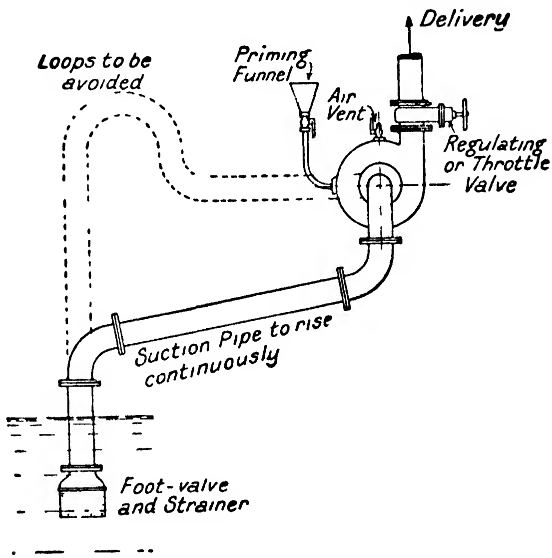


FIG. 198.—Elements of small pumping installation.

305. A Simple Layout. The elementary system shown schematically in Fig. 198 illustrates in a preliminary manner some of the points raised in Chapter XVIII. The *pump* might be any one of the units depicted in Figs. 38 or 39. In accordance with the rules of § 287, the *suction pipe* is short and rises continuously upwards ; a pipe having loops as shown by broken lines would only be admissible if the pump had self-priming characteristics, § 156, and would not be recommended even then. A combined *foot-valve and strainer*, § 290, keeps floating material

out of the system and prevents return flow when the pump is stopped. There is a priming funnel for admitting water to the casing, and an air vent for letting air out, § 291. The *sluice-valve* on the delivery side of the pump can be used either as an isolating valve or as a means of regulating the discharge, § 289.

306. Low-lift Plant. A good deal more consideration must be given to such a problem as is posed diagrammatically in Fig. 199; evidently some of the factors reviewed in § 297

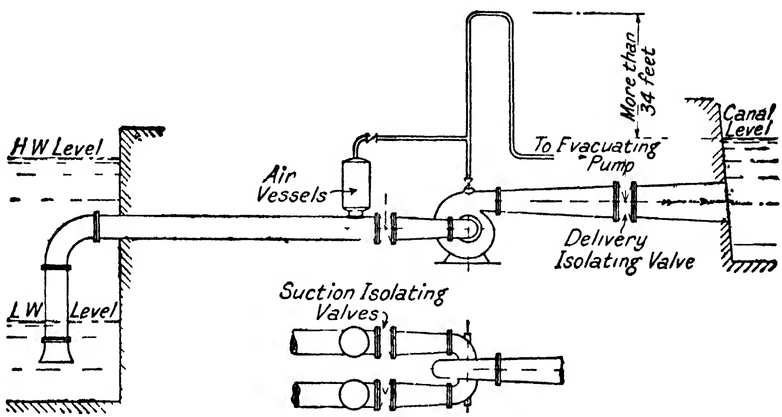


FIG. 199.—Typical low-lift centrifugal pumping plant.

are now operative. From a river whose surface level varies seasonally through a considerable range, water is to be drawn and it is then lifted into a canal which may be subject to minor variations in level. In this instance a horizontal centrifugal pump of medium specific speed has been chosen, and because of its fairly considerable size, the pump has twin inlet branches as in Fig. 39 (iv). Water rises directly into these branches through twin suction pipes which have bell-mouthed inlets unobstructed by foot-valve or strainer. Twin air vessels collect the small amounts of air that are continually released from solution as the pressure on the water is reduced; the air is periodically drawn off by the evacuating pump, § 287. Accidental flooding of this auxiliary pump is prevented by the long vertical loop in the pipe circuit; this expedient is even more effective than the float-valve mentioned in § 292. On the delivery side

of the main pump there is a diverging diffuser that ensures the maximum re-conversion of velocity energy, § 288.

In the general lay-out of the plant the controlling factor may often be the height above datum of the lowest water level in the river; for this figure, together with the value of the maximum permissible suction lift, § 253, establishes the vertical location of the pump axis. The diagram shows that in the present instance the pump must be set not only below ground level but below maximum river level. Two difficulties may thus be foreseen: during the construction of the plant there may be troublesome excavation in bad ground near the river bank,

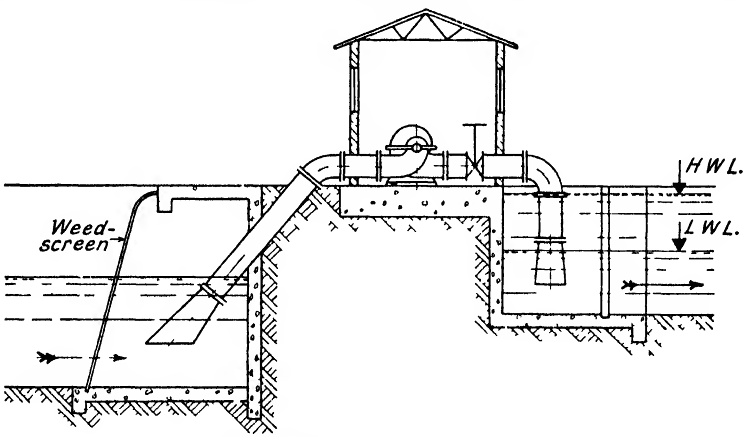


FIG 200—Centrifugal pump for land drainage scheme

and during operation an exceptionally high flood or some other mischance may drown the machinery in the pump chamber. If these risks are deemed to carry weight, it would be expedient to keep the pump chamber as small as possible, and to mount the motive unit up above, well out of reach of flood water. A heavy flat belt might transmit the drive from the motive unit—Diesel engine or electric motor—to the pump shaft. If, on the other hand, the risk of flooding was accepted and specially guarded against, then direct-coupled pumping sets would be preferable.

307. Land-drainage Plants. When the range of water level is relatively restricted, and the total lift does not exceed 10 or 15 ft., a simpler disposition is possible. Open conduits may bring the water up to the pump house, and carry it away

on the delivery side. Such an installation, still embodying centrifugal pumps, is illustrated in Fig. 200. If, as often happens, the water is discharged into a tidal stream, the major fluctuations in surface level now will naturally be on the outlet side.

Because the pump is here set above the highest level on the delivery side, it is said to work in a *siphonic circuit*. The corresponding advantages are explained in § 310. One small complication results. The pump is itself unable to feed sealing-water to the glands, § 85, and therefore a separate supply of pressure water must be contrived.

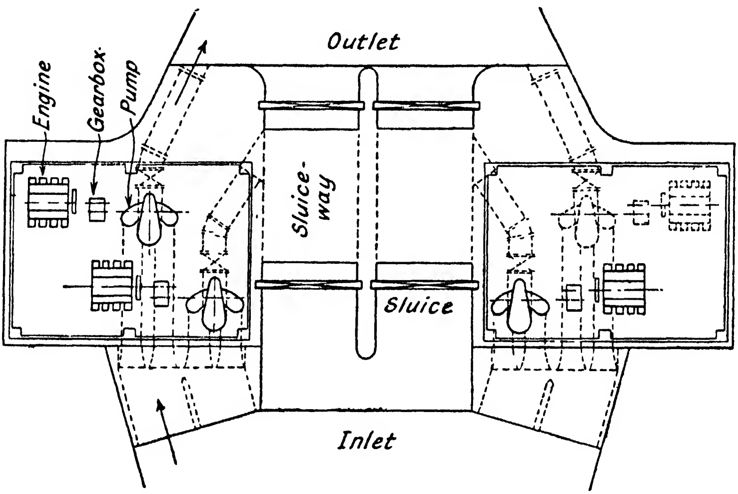


Fig. 201.—Centrifugal pumps for land-drainage, each delivering 800 tons per minute against 12 ft. head.

The plan view reproduced in Fig. 201 shows a fine example of a large drainage plant ; it makes clear how the civil engineering aspects of such a project may determine the choice of the mechanical equipment (*). This installation is interposed in a natural drainage channel that ordinarily carries a free or gravitational flow of water : the pumps, that is to say, are lying idle. They are only started up when the inlet water level rises above a certain limit, either because of heavy rainfall over the regions draining into the channel, or because of high tides affecting the outlet level. When this occurs the sluice-gates in the free-flow channel are closed, and the entire discharge is

diverted through the pumps. These quickly take command of the situation, drawing down the water on the suction side and heading it up on the delivery side, thus increasing the water slope in the channel sufficiently to pass the heavy flow.

Three main pumping units are accommodated in twin pump-houses, of which one lies on either side of the sluice-ways, Fig. 201. Space is also available for a fourth set. Like the pumps previously described, these machines are of the horizontal double-inlet centrifugal class ; the system of inlet and outlet conduits resembles that indicated in Fig. 199, except that now these large passages are formed beneath the pump-house floor. The motive units chosen have particular merits for their duty here ; the weight of the horizontally-opposed 8-cylinder oil

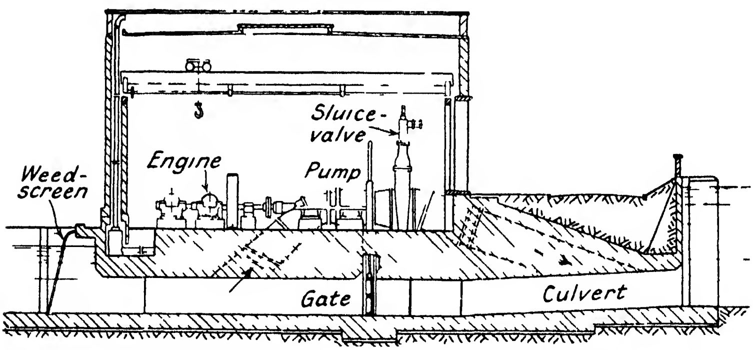


FIG. 202 —Half axial pump driven by oil engine.

engines is well distributed, a most desirable qualification when the entire buildings and their contents have to rest on unreliable ground. Although the engines are nominally slow-speed machines, it is nevertheless necessary to take the drive to the pump-shafts through step-down (speed-reducing) gear-boxes.

308. Screw-type Pumping-plant. Horizontal double-inlet centrifugal pumps, as embodied in the installations shown in Figs. 199, 200, and 201, are very reliable machines and they have earned the confidence of generations of engineers. Yet there are many occasions when the superior claims of screw-type pumps must be admitted. For handling large volumes of water the most adaptable machine in this category is the half-axial, Fig. 66, § 102. In Fig. 202 such a pump is seen direct-coupled to a two-cylinder horizontal oil engine, in such a manner

as to give a particularly easy run to the water (*). The conditions of service are identical with those described in § 307. The pump-house is built across a natural stream, which normally flows by gravity through culverts passing beneath the building; one of them is clearly shown in the diagram. When the inlet water level rises unduly, the culverts are closed by lowering the sluice-gates, and the pumps thereupon take charge. The inlet and outlet pump conduits communicate directly with the culverts.

Since the outlet water level may rise above the level of the pump, an isolating device on the delivery side is essential. In this particular installation each pump has a power-operated sluice-valve; its size in relation to the pump diameter is significant, suggesting that we would gladly dispense with so costly and cumbersome an item if it were anyhow possible.

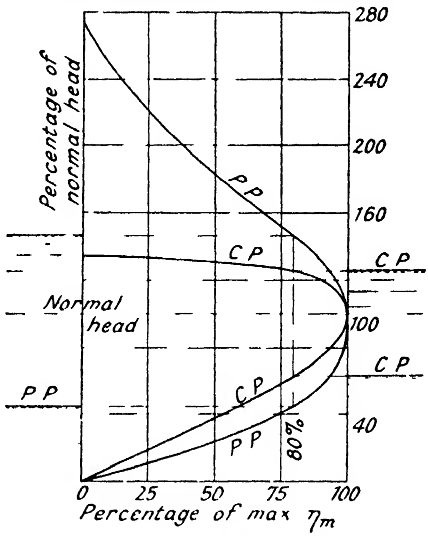


Fig 203 — Comparison between performance of centrifugal pump (C P.) and propeller pump (P P.)

309. Propeller-type Installations. For a given duty, the pure axial-flow pump is always more compact than any other, and it runs at a higher speed. Its characteristic performance is peculiarly favourable in the conditions that may now apply, viz., varying head and constant speed. If the information presented in Fig. 141, § 213, is rearranged in another way, we get the graphs of Fig. 203; efficiency at constant speed is plotted against head. From these curves it is evident that: (i) for a stated range of relative water levels, the *minimum* efficiency of the centrifugal pump is far lower than the corresponding efficiency of the axial-flow pump, or (ii) for a stipulated permissible drop in efficiency, the axial-flow pump can submit to a much wider

variation of head than the centrifugal pump can. It is the second type of comparison that is shown graphically in Fig. 203. The condition here, that the pump efficiently must not be allowed to fall below 80 per cent. of the maximum efficiency at the design point, imposes the following limits :— For the propeller pump, the variation in head is 105 per cent. of the normal head; for the centrifugal pump, the equivalent figure is only 65 per cent.

Nor is the upper part of the graph without significance. Although normally we should not like to penalise the axial-flow pump by running it in a zone of abnormal lift and reduced efficiency, yet there may be occasions when it would be extremely convenient to do so. Provided the motive unit will stand the temporary overload to which the shape of the pump power characteristic will inevitably subject it, § 213, it will be reassuring to know that the installation can meet these emergencies. The centrifugal pump certainly could not do so; as its graph in Fig. 203 indicates, the pump would be put out of action by a comparatively small increase in head.

(Example 38)

310. Disposition of Axial-flow Pumps. The simplest arrangement of propeller pump was shown schematically in Fig. 32, § 48; hardly any kind of pumping plant could conduct the water more directly from the lower to the upper level. When the diagram has been developed into a working drawing, it will be found that the layout still remains highly favourable for very low lifts. Both this disposition, and the one shown in Fig. 68, § 103, are manifestly suitable for direct-coupling between pump and vertical-spindle electric motor.

Other arrangements are illustrated in Fig. 204. The notable feature of the small motor-driven horizontal pump, (I), is that it is set in a *siphonic circuit.* Three important advantages result: (*a*) the bulky and costly delivery isolating valve is eliminated, § 308; (*b*) there is no reflux valve or its equivalent to cause dangerous "slam" pressures, § 273; (*c*) the pump is set above pump-house floor level, easily accessible for inspection or repair. In passing through the system, moreover, the water has the same easy run, free from abrupt changes of direction, that it has in the half-axial pump, Fig. 202. On the other hand, an evacuating pump is indispensable, § 292, and

an automatic vacuum-breaker must be provided on the pump casing. The purpose of this accessory, § 329, is to admit air into the casing when the pump stops and thus to prevent return flow of water and reverse rotation of the shaft. A siphonic disposition on the lines of Fig. 204 (I) is well adapted also to axial-flow pumps direct-driven by oil engines. **(Example 40)**

In the alternative arrangement shown in Fig. 204 (II), the water still has an easy path ; and furthermore, the inclined layout permits the pump to be permanently submerged, thereby eliminating priming apparatus and routine (*). Reflux flap-valves are now necessary to prevent return flow, and as these may be several feet in diameter they may "slam" very violently if the moving parts are not carefully controlled, § 330. When the pumps are as big as those depicted in diagram (II), having individual discharges up to 10 tons/sec., a step-down gear-box between electric motor and pump is obligatory. That is an important point : although the *relative* speed of propeller pumps is high, the *actual* speed in revolutions per minute may be quite low. Another point to be studied by the designer is . will the inclination of the various rotating shafts give rise to special problems of bearing lubrication ?

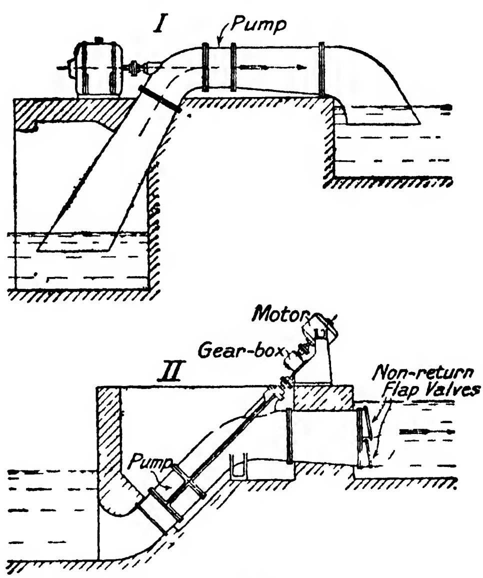


FIG. 204 —Horizontal and inclined axial flow pumps.

If the inclined propeller pump has adjustable rotor blades, as in Fig. 70, § 104, then very close and efficient regulation of the discharge is obtainable, § 244, and it may thereby be feasible to run the station with fewer units, § 300 (v).

Still other dispositions of propeller pumps are shown in Figs. 221 and 223.

311. Vertical-shaft Pumps. Although no rigid classification into horizontal-shaft pumps and vertical-shaft pumps has here been attempted, yet when framing provisional projects it is often prudent to compare the two types before making a final decision, § 300 (i). The vertical-shaft installation shown in Fig. 205 is meant for just the same kind of service as the pumps in Fig. 199, § 306, viz., varying discharge against low or medium, and varying, lift. Here also, as the lower key diagram indicates, the pumps are a long way below river water level at times of high flood ; but now the driving motors are lifted high above danger (*). Another feature common to the two installations is the tapering outlet diffuser pipe.

The side-inlet type of mixed-flow centrifugal pump chosen for the vertical-shaft plant is the one illustrated in Fig. 61, § 99 ; it is the type which above all others gives the promise of high efficiency. A long, wide-bore suction pipe, supported on a staging carried out into the river, brings water to each pump, and as in Fig. 199 the outlet pipes of the pumps deliver the water to the head of a canal. Step-down gear-boxes are interposed between the electric motors and the pumps. Constant-speed motors would manifestly be unsuitable if the installation had to deal with the entire range of water levels indicated in the key diagram, Fig. 205, but in fact the pumps are out of action during the flood stage of the river. Even for the smaller variations of levels actually imposed upon the pumps, two-speed motors would be very advantageous, § 278 (ii).

Very large vertical-shaft pumps are described in § 336.

312. Other Vertical-shaft Installations. Another solution of the low-head pumping problem is illustrated in Fig. 206, and as it embodies oil engines it may profitably be compared with the scheme described in § 308. Of the medium-speed vertical type, the engines drive the pumps through step-down bevel gear-boxes. A novel point about the layout is this : not only are the inlet and outlet conduits formed within the concrete foundations of the pump-house, but the volute casings of the pumps are similarly constructed (*). As compared with the normal type of pump as used, for example, in the plant shown in Fig. 201, there is consequently a substantial saving

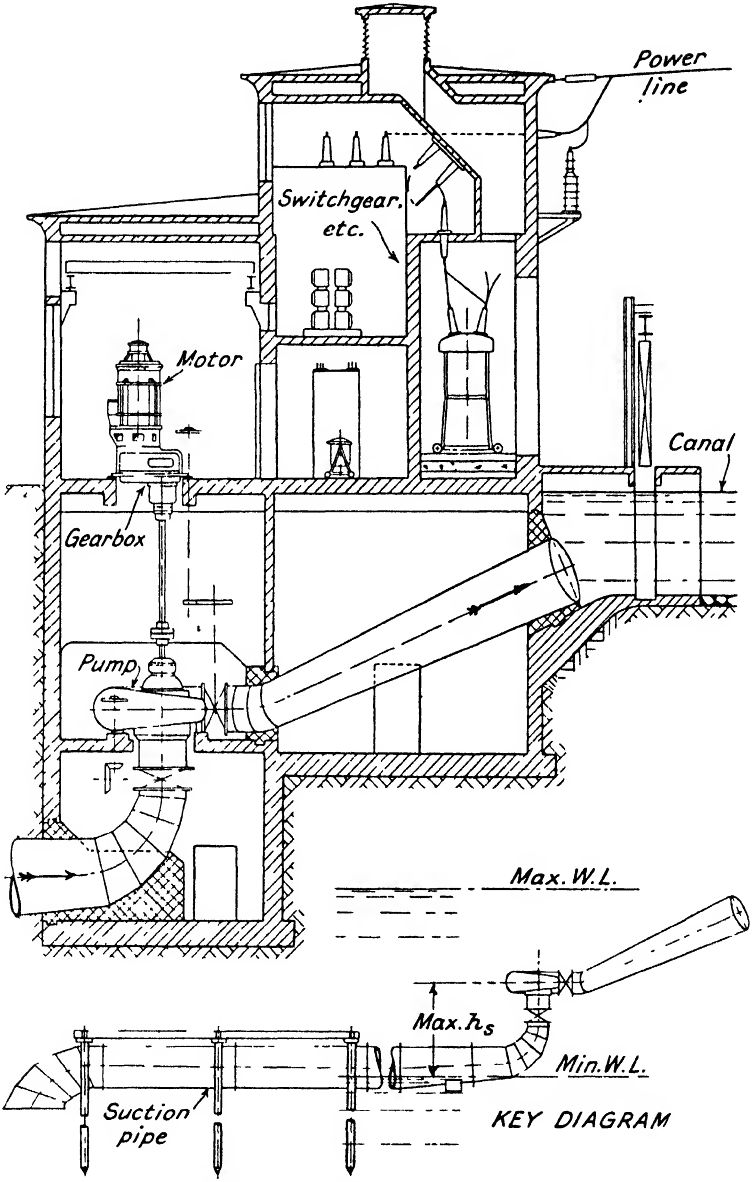


FIG. 205.—Vertical-shaft centrifugal pumps in irrigation pumping-plant.

in weight of iron castings. Moreover, the machine-house floor is left relatively free and unobstructed, and the path of the water through the system is as direct as is feasible with this class of pump—the mixed-flow side-inlet centrifugal type. As Fig. 206 suggests, an installation of this size cannot escape the burden of costly sluice-gates. In regard to the gravity-flow passages

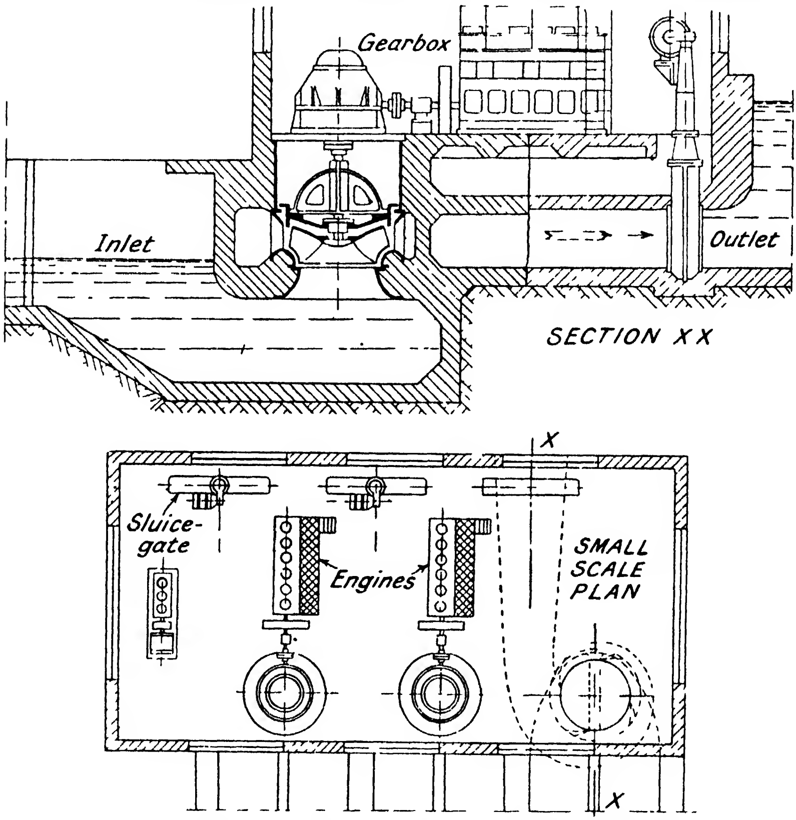


FIG. 206.—Low-lift centrifugal pumping-plant with "built-in" volutes.

shown in Figs. 201 and 202, the corresponding channel for the plant now in question passes outside the pump-house and is not seen in Fig. 206.

In the vertical-shaft scheme depicted in Fig. 207, the driving-shafts between electric motors and pumps are so long that intermediate steady-bearings are needed to support them. There is a special object in setting the pumps below the suction

surface level. They are destined for lifting screened sewage from a collecting pit into an upper channel, and if the liquid flows freely into the pump intakes there is a reduced risk of accumulations of solid and semi-solid material at the bottom of the suction chamber (*). Naturally this involves a good deal of additional excavation as compared with what would be required if the pumps were mounted above suction level, but the expense is found to be justified. Besides, the pumps now need no priming devices

To ensure that the nominally "dry" underground pump chamber remains dry, reliable auxiliary pumps are essential, § 293; the one shown in Fig. 207 is of the single-stage bore-hole type. Whether the main side-inlet centrifugal pumps should be of virtually standard construction, or whether they should be modified as described in § 149, is manifestly

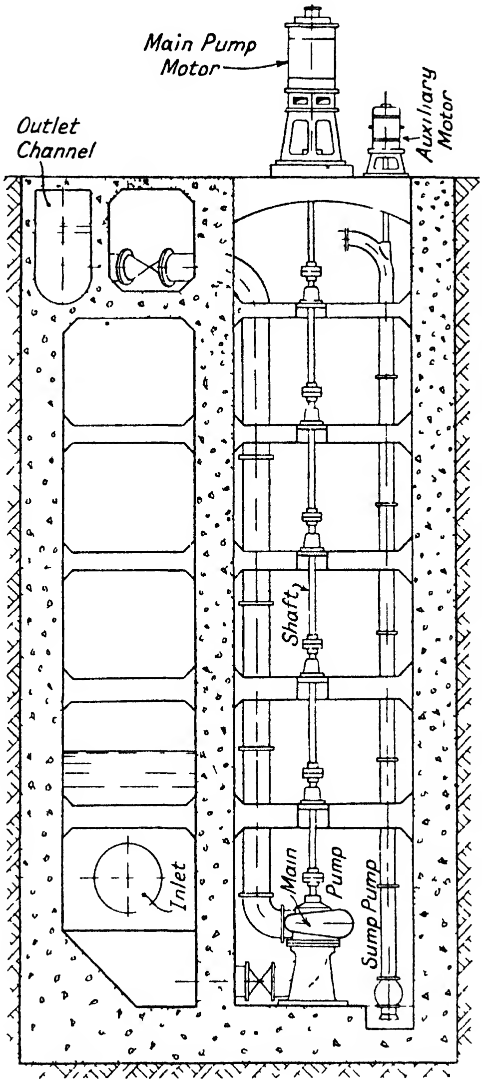


FIG. 207.—Vertical shaft centrifugal pumps in sewage-disposal works.

a matter that will depend upon the character of the sewage.

313. Borehole Pumping-Plant. The limiting example of the vertical-shaft installation is provided by the borehole plant, in which the transmission shaft between motive unit and pump may be several hundred feet long; this shaft is now carried axially inside the rising main or delivery pipe, as explained in § 132. A simple type of borehole scheme was shown in Fig. 89; more complete information is given in Fig. 208. Here are shown two sets of pumps, each set comprising a driving motor, a borehole pump at the bottom of the borehole, and a force-pump or booster pump at ground-level (*). The *borehole pump* itself is of the general construction explained in § 132, and its design is controlled by the rules of § 137. The head it generates is sufficient only to feed the water to the force-pump, which then takes charge and delivers to the water the balance of the energy.

This system of sharing the load, though not obligatory, has various advantages; the less the burden placed upon the lower pump, the lighter in every sense can be the equipment working below ground. Moreover, as the diameter of the force-pump is not restricted in any way, it can be proportioned with the chief aim of securing maximum efficiency. If for any reason it is inconvenient to mount the two pumps on the same shaft and drive them by the same motive unit, it is quite easy to use as a force pump a standard horizontal machine disposed on the pump-house floor. This disposition may have merits of its own. At times of low water in the borehole, both pumps would be kept running; at times of high water, the force pump could stand idle and the borehole pump could comfortably do all the work. **(Example 41)**

Fig. 208 shows various features usually found in borehole pumping-plants. The delivery pipes from the two force-pumps unite to form a single rising main which delivers the water to its distant destination; this main has an air-chamber which helps to reduce water-hammer when the output of the station varies, § 335, and a Venturi meter for flow measurement, § 294. The heavy construction of the travelling-crane is very noticeable, and is essential because of the dead load to be lifted when dismantling a pumping-set.

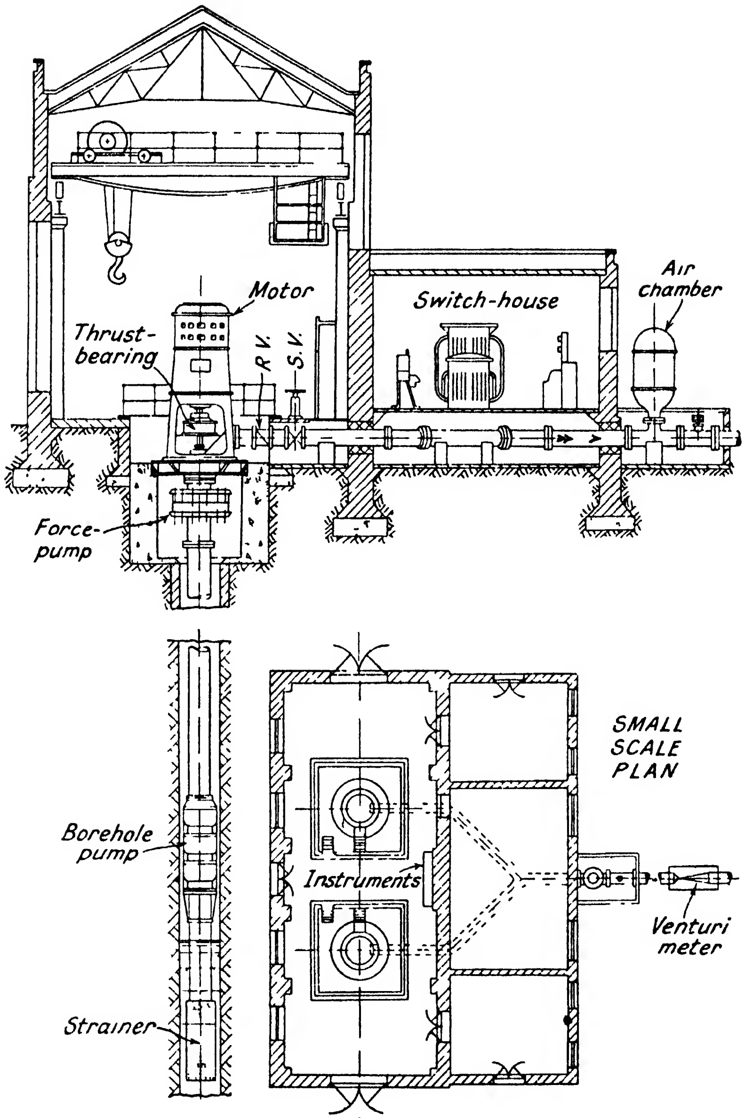


FIG. 208.—Borehole pumping-plant.

314. Details of Borehole Installations. Taking over the description of details from the point reached in § 137, we may note the following points (*) as we accompany the water upwards :—

Strainer. A strainer for keeping out sand is seen in Fig. 208, but some authorities prefer to do without one.

Foot-valve. In the use of this item, § 290, there are likewise differences of opinion. Submersible borehole sets are preferably fitted with a foot-valve, § 136, but in shaft-driven sets there is much to be said for using only a reflux-valve at ground-level, as in Fig. 208. When such a set stops, the water contained in the system below the reflux-valve runs back into the well, driving the shaft round in the reverse direction, and in this way the pump passages and the inlet strainer may be at least partially cleared of accumulations of sand. If, on the other hand, the water is retained in the system by a foot-valve, this additional weight must be lifted when the time comes to withdraw the pump from the well.

315. The Thrust-bearing. As explained in § 132, this important component has to sustain the entire weight of all the revolving parts that lie below the half-couplings of the motive-unit, viz., driving shaft and all its couplings, and rotors of borehole pump and of force-pump. In addition, it must withstand the net hydraulic thrust on the pump rotors, § 123. Frequently the axial thrusts of the two pumps both act in the same downward direction, but it is sometimes possible to reverse the aspect of the force-pump impellers so that their thrust tends to neutralise that of the borehole pump. The elastic extension of the driving shaft under the gross load may be sufficient to disturb quite appreciably the axial alignment of the borehole pump impellers in relation to their diffusers, § 75 (*c*), and the thrust-bearing should therefore include provision for vertical adjustment.

The customary position for the thrust-bearing is just above the water collecting-piece, as in Fig. 208 ; between the bearing and the motive unit is the main flexible coupling. As the bearing will be of the tilting-pad type (Michell or Kingsbury), cooling arrangements must be provided for it, which in large systems may be fairly elaborate. If the installation is designed

for reverse rotation after stoppage of the pump, the thrust-bearing should be modified accordingly.

It will be understood that the disposition just described expressly isolates the rotating element of the driving-motor from any vertical external forces; the motor thrust-bearing only has to take care of thrusts arising within the motor. In small sets, however, a combined arrangement is feasible. The motor shaft is hollow: the pump shaft is carried upwards through it: the two are locked together by adjusting nuts at the upper end, where we find the single heavy-pattern ball-bearing which supports the *entire* downward thrust and transmits it to the crown of the motor frame.

316. Disposition and Capacity of Motive-unit. The particular conditions of service of borehole pumps, § 242, justify the engineer in devoting special study to the choice of motive-unit. Wide fluctuations in the well water-level may impose varying heads on the pumps, which implies the desirability of varying pump speeds; and as the pumps will probably run continuously at a fairly high power output, expensive systems of speed regulation may be found thoroughly worth while (*). According to the size of the installation, any of the variable-speed *electric motors* described in §§ 277, 278 can be recommended. Variable-speed *oil engines* may drive the pumps through step-up bevel gear-boxes, § 285; for variable-speed vertical shaft *steam turbines*, step-down gear-boxes are required.

In specifying the power output of the motive unit for a borehole pump, there are special allowances to be made. First of all, the effective head on the pump must be computed so as to include the energy loss as the water ascends to the ground surface; the frictional loss in the vertical pipe will be augmented by the eddy loss as the water flows past the spiders that support the steady bearings. Secondly, the power to be fed into the coupling at the top of the driving shaft must include the total S.H.P. input required by the pumps, *plus* a supplement which represents transmission power losses. These transmission losses consist of (i) power loss in the steady bearings (corresponding to the mechanical power loss P_b in the pump itself, § 188), (ii) surface friction loss as a result of contact between the revolving shaft and the water in the vertical pipe

(corresponding to the disc friction power loss P_d in the pump, § 189).

A tentative estimate of this complex total may be arrived at thus :—

If P_{sh} denotes total transmission power loss, as defined above,

L ,, length of shaft in feet,

D ,, diameter of shaft in inches,

N ,, speed of shaft in r.p.m.,

then for a range of speeds of about 1000 to 1600 r.p.m.,

$$P_{sh} - 0{\cdot}4 \left(\frac{L}{100}\right) (D)^{2.1} \left(\frac{N}{1000}\right)^{2.6}$$

CHAPTER XX

SOME PARTICULAR INSTALLATION PROBLEMS

317. Description of the Problems. In this chapter the intention is not so much to examine typical pumping-plants as to review particular aspects of the installation, noting what varied solutions can be found for a specific problem. While some of the plants described may be of quite a general nature, others may be expressly designed for a closely-defined duty. In one way, then, the descriptions will serve as a commentary on the material contained in Chapter XVIII; in another way they are a continuation of the treatment begun in Chapter XIX.

The following sub-headings will indicate the nature of the ground to be covered :—

(i) Grouping of pumping units.
(ii) Boosting systems.
(iii) Systems for controlling pressure-surges in conduits.
(iv) Special priming systems.
(v) Systems of auxiliary pumping plant in steam power stations.
(vi) Transportable pumping plants.
(vii) Hydraulic storage plants.

Grouping of Pumping Units

318. Types of Grouping. In all pumping problems the fundamental need is to deliver to the flowing liquid a specified amount of energy. This energy can either be given in a single "shot" or dose, or it can be divided into portions and supplied to the liquid at a number of points along the conduit. Translated into practical language, this means: if a stated quantity of liquid is to be moved from point *A* to point *B*, shall we use one rotor only or must we use a number of rotors? If more than one rotor is required, how can these multiple rotors conveniently be disposed? There are at least four possibilities:—

(i) The rotors can be assembled within a single casing, working either in parallel or in series. This solution was fully described in Chapter IX.

(ii) We may use two or more complete pumps, each pump having its own motive unit, and these pumping-sets can be arranged in the pumping-station in series or in parallel.

(iii) Two or more pumps may be permanently coupled to a single motive unit, the whole thus constituting another kind of pumping-set.

(iv) When the liquid must be moved through a distance of many miles, a series of pumping plants must be spaced along the pipe line, each plant itself comprising several pumping-sets.

These dispositions will respectively be denominated thus: (i) Multiple-rotor *pump*, (ii) Single-pump *sets*, (iii) Multiple-pump *set*, (iv) Multiple-*station* system.

319. Single-pump Sets. This is the usual system of grouping pumps and motive units, and has already frequently been mentioned. Here are some considerations to be kept in mind when fixing the optimum *number* of sets that will give the stipulated duty: Other things being equal, a single large pump will not only be cheapest to buy, § 298, but it will have a higher efficiency at its design point than any smaller pump of the same shape could attain. On the other hand, a group of small pumps working in parallel can be manipulated so as to give a much more favourable part-flow efficiency, §§ 243, 244; and in any event the necessity of providing stand-by plant rules out the possibility of relying upon one pump only. Still greater flexibility in operation can be achieved if the pumps are

of two different *sizes*. Assuming that the individual capacities are in the ratio ($\frac{1}{2}$) : (1), then the gross station output can have the relative values $\frac{1}{2}$, 1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$, etc. In stations equipped with constant-speed pumps, e.g. those illustrated in Figs. 199 and 205, this way of regulating the discharge is not only convenient but essential—it is the *only* practicable means of adjustment. The only material drawback is that two types of spare parts must be kept in the station. If, for this reason, only one type of pump can be allowed, there still remains a method of securing an infinitely-variable rate of discharge. Of the total number of identical pumps in the station, all but one can be constant speed machines; the remaining machine can be given speed-variation, and by this means it can take care of all minor changes in gross discharge.

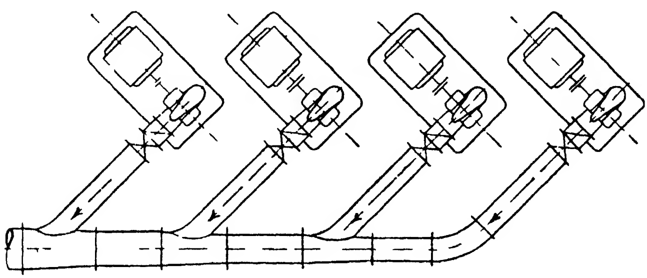


FIG 209 Multiple pumps set in echelon.

Usually the individual pumping sets are disposed side by side in the pump-house, as in Figs. 202, 206. A staggered arrangement is seen in Fig. 201, while in the accompanying Fig. 209 the pumps are set in echelon formation; not only are the units easily accessible, but the connections to the outlet manifold are as direct as possible, § 288 (i). It is here assumed, Fig. 209, that the suction pipes rise vertically upwards to the pumps. The installation shown in Fig. 210 is remarkable for the great complexity of the conduits (*). It is typical of what may be required in certain kinds of dock-pumping equipment, where it may be necessary either to pump water from the dock into the sea, or from the sea into the dock (*). Each large vertical-shaft motor-driven centrifugal pump has its own reflux valve and four electrically-operated sluice-valves, which permit the water to be directed as desired.

When two pumping-sets have to be operated either in series or in parallel, § 243 (iii), they can conveniently be arranged as in Fig. 211. Diagram (i) shows the two pumps working in

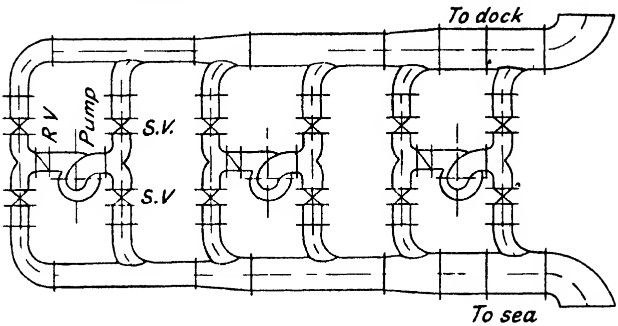


FIG. 210.—Arrangement of pumps and conduits, etc., in dock pumping plant.

series, while in diagram (ii) they are in parallel. The change-over is effected by suitably manipulating the valves (*a*), (*b*), and (*c*). To enable one pump only to remain in service, the isolating valves of its fellow would be closed.

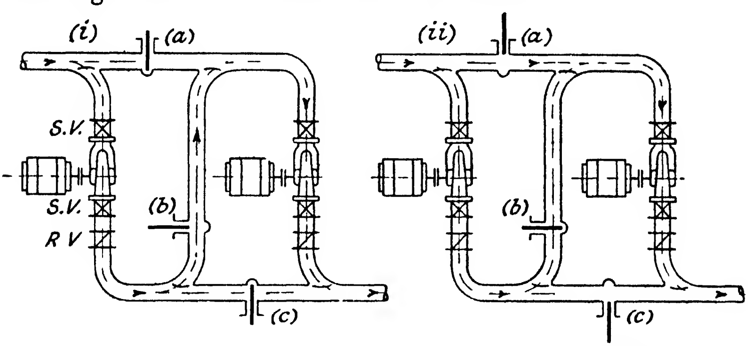


FIG. 211.—Pipe system for pumps disposed in series or parallel.

320. Multiple-pump Sets. (*a*) *Pumps in Parallel.* The starting-point in considering this arrangement is the twin-rotor pump described in § 116. There, the two rotors were accommodated in a single casing, Fig. 77; now, we provide two standard centrifugal pumps, and one motive unit to drive both, Fig. 212 (*a*). The multiple-pump set behaves in all respects as a single-pump set of high specific speed; the shafts are permanently coupled together and so are the pipes (*). The combination is more economical in every way: as compared with

the two equivalent single-pump sets, there is an economy of space, a saving in first cost, and, because of the larger size of the motive unit, a slight saving in power consumption. Although it is true that this large motive unit might be set to drive a single large screw-pump or propeller-pump, such a solution might be ruled out by reason of the suction lift conditions, § 253.

(*b*) *Pumps in Series.* Already we have seen the advantages of using two pumps in series (*), driven by a common motive unit; this arrangement, as illustrated in Fig. 208, § 313, is frequently adopted in borehole pumping plants. A system that is advantageous for horizontal-shaft sets is seen in Fig. 212 (*b*); on either side of a double-ended electric motor there is a direct-coupled multi-stage pump. Standard pumps having say, six or eight stages each could here be used, whereas if all the work

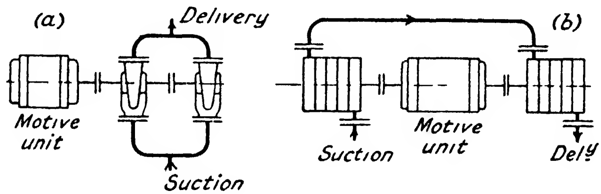


FIG. 212. —Multiple-pump sets.

had to be done by a single pump there might be serious constructional difficulties. Even as it is, there is one item of the high-pressure pump in Fig. 212 (*b*) that may need special treatment: it is the stuffing-box on the inlet side, §§ 129, 142, which is now exposed to the full delivery pressure of the low-pressure pump.

Although twin multi-stage pumps are shown in Fig. 212 (*b*), the system of coupling in tandem is equally applicable to single-stage pumps.

321. Other Multiple-pump Sets. (i) *Pumps having Different Duties.* It is sometimes worth while permanently coupling a set of pumps to a single motive unit, even though the pumps have different duties and are connected to different circuits (*). In a large waterworks, for instance, where a steam-turbine drives the pumps, one of the pumps may feed a low-pressure system while the other feeds a quite independent high-pressure system; the only thing the pumps have in common is

the same rotational speed. Another example may be found in a steam power station, in which perhaps the circulating pump and the extraction pump receive their energy from a single motive unit.

(ii) *Pumps running at Different Speeds.* Here the individual pumps depending on a single motive unit are sharing the same duty, but for one reason or another they run at different speeds. Examples are: (*a*) The suction lift conditions may involve the use of a low-pressure pump whose function is merely to feed the water into the main or high-pressure pump, § 255 (iii) (*c*). Because of the high *total* head, the main pump must run very fast: because of the high *suction* head, the low-pressure pump must run relatively slowly. The difference between the speeds could be bridged by a step-down gear-box, Fig. 213 (*a*).

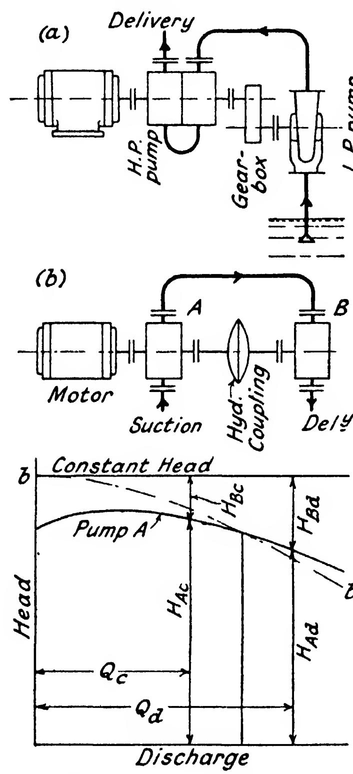


FIG 213.—Other multiple-pump systems.

(Example 42)

(*b*) The method of discharge regulation by means of a hydraulic coupling, §§ 284, 300 (iii), is capable of refinement if two pumps are used instead of one. The complete pumping set then consists of: first, the constant speed electric motor; second, the main high-pressure pump, *A*, direct-coupled to the motor; third, the hydraulic coupling; fourth, the low-pressure variable-speed pump, *B*, Fig. 213 (*b*); the whole being designed to deliver a varying discharge against an assumed constant head. Maximum discharge is given by running the coupling full of oil, which implies minimum slip and maximum speed of the low-pressure pump. To reduce the discharge, the slip is increased, pump *B* is slowed down, and the head generated by

this low-pressure pump is correspondingly reduced. As this throws a greater proportion of the total head on to the high-pressure pump, the flow through the set must decline as dictated by the high-pressure pump characteristic.

The lower diagram in Fig. 213 (*b*) show how the load is shared under conditions of low discharge, Q_c, and of high discharge, Q_d. The corresponding heads contributed by the two pumps are H_{Ac} and H_{Bc} in the one case, and H_{Ad} and H_{Bd} in the other. If the broken line *bb* (representing the parabola

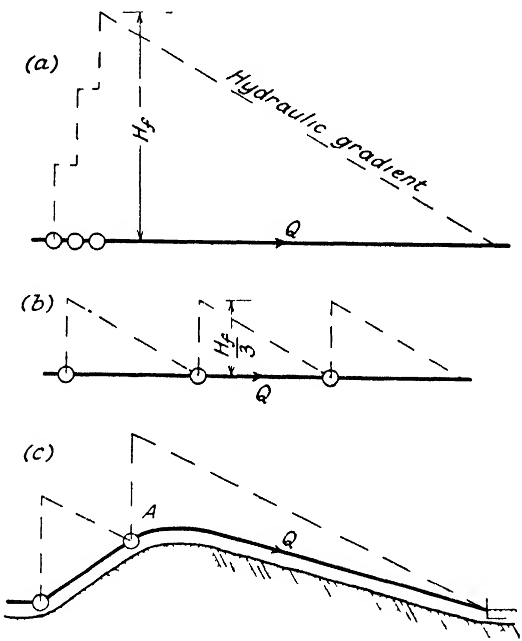


FIG. 214.—Pumping stations for pipe lines.

of maximum efficiency for the variable-speed pump, § 220) could coincide with the full-line characteristic for pump A, then losses in pump B would be at their minimum. In any event the energy loss in the hydraulic coupling would be less than in the very much larger coupling required in a single-pump set.

322. Multiple-station Systems. According to the definition of § 318, we are now dealing with a series of pumping installations spaced along a pipe-line (*). Considering the

simplest case of a long horizontal pipe in which the external head is wholly frictional, Fig. 214, there would at first appear to be a choice between (*a*) concentrating the entire pumping plant at the head of the line, or (*b*) disposing groups of pumps uniformly at equal distances apart. As the diagrams show, the total energy input to the liquid must be the same in either case; but the hydraulic conditions in the pipe-line are very different. The maximum pressure in system (*a*) would be three times as great as in system (*b*), and this difference would undoubtedly be reflected in the cost of the pump and pipe-line. It is true that the overall cost of installing and running the individual *stations* might be higher than the corresponding figure for the combined station; yet there are operating advantages to be taken into account also (*). Storage reservoirs at each station will ensure greater flexibility of working, and less interruption to the service, if a pipe should burst or a pumping set fall accidentally out of commission. **(Example 43)**

Uniformly-pitched pumping stations are hardly likely to provide the best solution if the pipe-track follows undulating ground. The location of the plants will doubtless be governed by (i) the necessity of avoiding negative or suction pressures anywhere in the main pipe-line, (ii) the desirability of keeping the positive pressure within reasonable limits, (iii) the possibility of utilising local sources of power, of water, and of supplies for the operating staff. If, for example, the profile of the country were of the general nature shown in Fig. 214 (*c*), and a single intermediate station were specified, then it should manifestly be located at *A* rather than mid-way along the line.

Boosting Stations

323. Principles of Boosting. In general, a booster pump is one that augments or "boosts" the pressure of the liquid already flowing through a hydraulic system; it does not itself initiate the flow of liquid, but only helps the liquid along. In this broad sense, the installations shown in Figs. 201 and 202 are undoubtedly carrying out a boosting operation, for by generating an additional head they increase the speed of the liquid along its original channel. The uppermost of the two pumps in the borehole set described in § 313 could be termed a

booster pump because it imparts additional energy to the water fed to it by the lower pump.

The term "boosting station", on the other hand, has rather a particular significance. It is applied to pumping installations interposed in the water mains of a water-supply undertaking, for the purpose of increasing the discharge in the mains. The elements of the problem are represented schematically in Fig. 215, where the boosting station is indicated by a pump set in a shunt circuit, arranged in parallel with a reflux valve in the main itself. Until the pump is started up, the flow along the pipe is dependent entirely upon the gravitational head H, i.e., upon the difference in level between the two reservoirs. But when the pump is running it generates a head

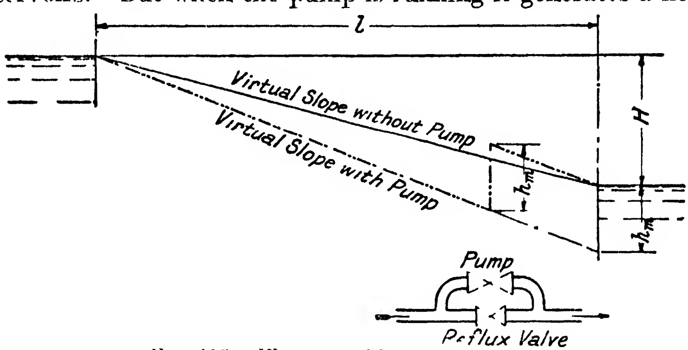


FIG. 215.—Elements of boosting system.

h_m which creates a "step" in the hydraulic gradient as shown in the diagram, with the result that the reflux valve closes and the whole flow in the system is now by-passed through the pump (*). That is an important point: no matter how small an additional head the pump develops, the pump has to carry the *entire* augmented flow. To what degree is the flow increased? That can be gauged by the increased slope of the hydraulic gradient: under the gravitational head H, the discharge was proportional to $\sqrt{\text{virtual slope}}$, or to $\sqrt{\frac{H}{l}}$; under the augmented head $(H + h_m)$, the boosted flow is proportional to $\sqrt{\frac{H + h_m}{l}}$. The effect of the pump, in short, is just as though the level of the lower reservoir had fallen by a distance h_m.

324. Types of Boosting Station, (i) *Without Water*

Storage. For use with long gravitational mains working on the hydraulic principle of Fig. 215, a suitable type of boosting installation is shown in Fig. 216. It comprises three standard

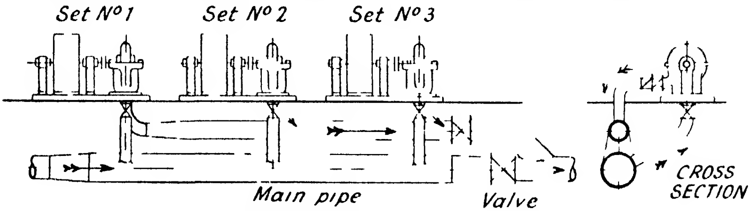


FIG. 216.—Arrangement of boosting plant.

centrifugal pumps, each driven by a direct-current variable-speed motor which takes its supply from special motor-generator sets ; for only in this way can the necessary close regulation of discharge be guaranteed (*). In place of the automatic reflux valve used in Fig. 215, there is now a power-operated valve in the main pipe which serves to divert the flow through the pumps when the gravitational head is insufficient. **(Example 44)**

From the graphs representing the internal and external heads on the pumps, Fig. 217, it is evident that if only a small increase in discharge is required, the pumps will be running under very unfavourable conditions: they generate a disproportionately small head in relation to the discharge. Ideally, therefore, the best routine that would keep the pump performance nearest to the ridge of maximum efficiency, § 222, would call for three pumps all working together for small degrees of boost, but only two pumps when the speed and discharge have risen considerably. Paradoxical though this may seem, the procedure is justified by the diagrams.

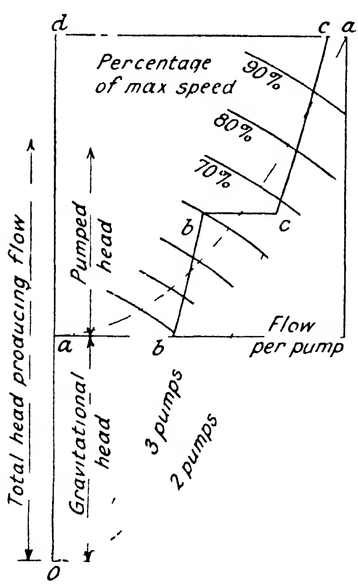


FIG. 217.—Optimum method of operating booster plant.

These graphs are plotted on the same basis as those in Fig. 166 (ii). But there are two differences in detail. Whereas in Fig. 166, the gravitational head H_s was plotted *above* the base-line, in Fig. 217 the head *aO* is necessarily laid off *downwards*. Moreover, the rates of flow or discharge in the booster diagram are expressed in flows *per pump*. As the total flow in the main pipe progressively rises, the sequence of operations would be as follows : Boosting would begin when the gravitational head *aO* was exhausted. Then three pumps would be set to work, each running at about 30 per cent. of full speed and each delivering a quantity *ab*. As the demand increased, the pumps would likewise be accelerated up to 60 per cent. of full speed. Thereafter one pump would be shut down, and the two remaining units would be still further speeded up, until the maximum flow in the pipe ($= 2 \times dc$) was attained, at very nearly full pump output. In this way the operating line *bbcc* is kept reasonably close to the pump maximum efficiency line *aa*. In point of fact it is unlikely that any station would run with so high a relative degree of boost ; a pumping head amounting to 125 per cent. of the gravitational head is needed to increase the pipe flow by only 50 per cent. This might imply prohibitive power costs.

In practice, in order to eliminate manual regulation of the pumping units and to enable automatic control to be enforced, § 281, it would be preferable to run the three pumps always as a group, with speeds rising and falling in unison. Alternatively, two pumps arranged *in series* might be advantageous, set in the kind of circuit illustrated in Fig. 211. Either of the two pumps would serve for small boosts, and both pumps for high boosts.

The location of the boosting station in the pipe-line is a matter of some importance because of its effect on the hydraulic gradient, Fig. 215. Evidently it will be essential to verify that the absolute pressure at any point in the pipe does not fall below atmospheric pressure, or rise above the safe capacity of the pipe.

325. Types of Boosting Station, (ii) *With Water Storage.* Small boosting installations are often designed so as to be completely automatic and unattended. They can then be arranged in some such manner as in Fig. 218. The storage chamber,

which may take the form of a cylindrical steel shell working under compressed-air pressure, serves two useful functions: (i) it permits the pump to work intermittently, thus ensuring a higher average efficiency, § 241; (ii) it cushions water-hammer effects in the main that might arise when the pump is started or stopped. Assuming that the station has been established to maintain adequate pressure in some outlying or elevated district of a water authority's area, let us now follow its working throughout a typical day.

During the early hours of the morning the demand for water is so small that the pressure gradient in the main pipe rises high above minimum levels, and the booster pump remains dead. Later on, there is an increasing draw on the supply,

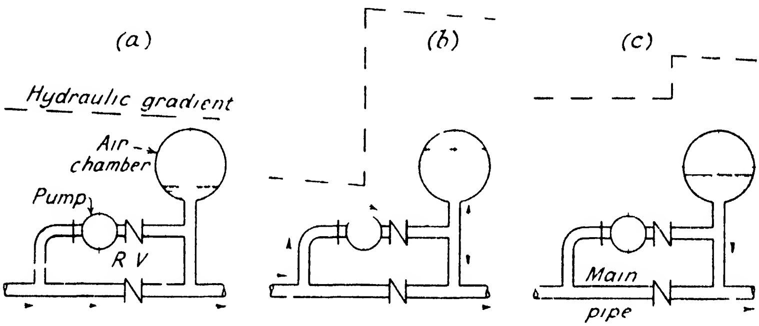


FIG. 218 —Booster plant with storage

which pulls down the pressure nearly to its lower limit. Meantime, as the air-pressure in the storage chamber has been exactly keeping pace with these changes, the stored water there has gradually been used up, Fig. 218 (*a*), as a result of the expansion of the air. At this point an automatic pressure-switch, § 281, starts the constant-speed motor It drives the booster pump at a rate *greater* than is required to supply the maximum demand on the system, and boosting at once begins as in § 324; but now its effect is modified by the air chamber, diagram (*b*). The air-pressure— and with it the pressure in the main can only slowly increase, as water accumulates in the chamber. This stored water, of course, represents the surplus delivered by the pump, above what is momentarily asked for by the consumers.

After perhaps half-an-hour or so, the storage chamber is again fully charged and the pressure there and in the main has

been restored nearly to its original peak value. The "high" contact of the automatic pressure-switch thereupon stops the motor, and for the next period the consumers are fed entirely by the reserve water which is forced out of the storage chamber by the compressed air, Fig. 218 (*c*). Again the pressure touches its minimum : again the booster pump is switched in : and so the cycle continues. Thus the respective lengths of the "storing" and "discharging" periods will be governed by the average demand on the system. Towards the end of the 24-hour day, we may expect that the draw on the system in general has slackened so much that gravitational flow is again resumed and the boosting plant thereupon shuts itself down for the night.

Control of Pressure Surges

326. Character of Pressure Fluctuations. It was explained in Chapter XVII that the starting or stopping of a pump may set up pressure waves that may be propagated throughout the whole piping system (*). If these are not properly controlled they may endanger the installation in at least three ways : (i) An excessive *negative* surge might cause the pipe to collapse inwards because of the unbalanced external pressure of the atmosphere, § 264. If the liquid column has been ruptured, the pipe may burst through excessive *internal* shock pressure when the two parts of the column re-unite. (ii) An excessive *positive* surge may likewise damage the pump or piping as a result of abnormal internal pressure. (iii) Violent slam-pressures may be generated by the closing of an automatic reflux valve, § 273.

An insight into the kind of preventive measures available was given in § 274. Before making a more detailed review, it is well to realise the relative importance of true surge pressures and of slam-pressures. The former can only reach a significant intensity when the delivery pipe is at least several hundred yards long, for it is only then that the pressure wave has time to build up (*). But slam-pressures can be quite dangerous when the whole hydraulic system is only a dozen yards long ; moreover, they are applied to the pump and pipe system so nearly instantaneously that their precise value is most difficult to estimate.

327. Inertia Pressures when starting the Pump. The reasoning of § 262 showed that the intensity of positive surge pressure when the pump is started from rest can never exceed a figure equivalent to the zero-discharge head of the pump. Yet for a *propeller-pump* installation even that might be regarded as excessive. The safeguard recommended in § 280—the use of starting-gear that brings the pump gradually up to speed—rules out the attractive simplicity of the direct-on-line squirrel-cage motor. Alternative expedients are sketched in Fig. 219. In diagram (i), there is a hand-operated by-pass which is opened before the pump is started and then gradually closed after the pump has picked up speed. The system shown in diagram (ii) is completely automatic and consists only of an overflow pipe, which positively limits the pressure during the starting period. It will be realised that such devices are intended to protect the driving motor rather than the pipes and pump, for expressed in pounds per square inch the worst possible pressures are only trivial.

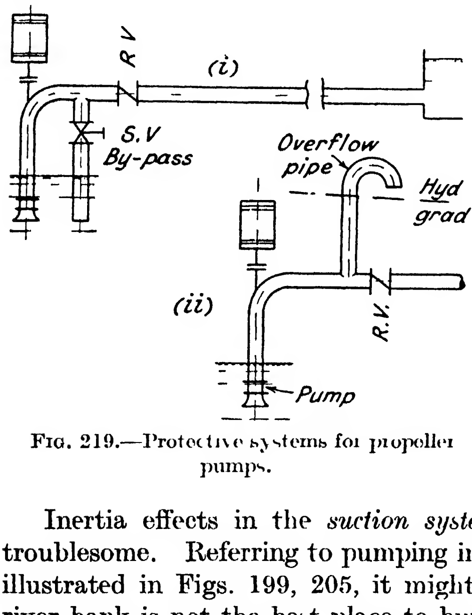


FIG. 219.—Protective systems for propeller pumps.

Inertia effects in the *suction system* may occasionally be troublesome. Referring to pumping installations such as those illustrated in Figs. 199, 205, it might easily happen that the river bank is not the best place to build the pump house. In order to find better foundations or better access from road or railway, the station may have to be set back some hundreds of yards from the river, as in Fig. 220, the water being taken to it through an underground pipe which terminates in a well. The individual suction pipes of the pumps draw directly from this well. What will happen when a pumping-set is started? Before the water column in the long underground pipe can be accelerated sufficiently, the water level in the well may be

drawn down several feet; and if the pump motors are controlled by immersed electrodes, § 281, the result may be that the motors are automatically switched off again. A series of starts and stops may follow before conditions are finally stabilised and steady flow has been established, thus overtaxing the starting equipment and motors. The remedy is to make sure that the suction well has a generous surface area.

328. Systems for Minimising Slam-Pressures, (i) *In Low-head Plants.* The only infallible method of eliminating slam-pressure when the pump stops is to leave nothing in the pipe that can possibly slam. But this condition has been fulfilled in various installations already illustrated, e.g. those in Figs. 199-202, 205-206. What need is there, then, for any further study of the problem? There are two reasons: (*a*) all these plants depend upon the presence of an alert attendant who can shut the sluice-valves or gates if power is accidentally

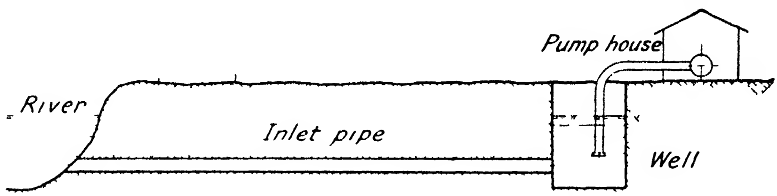


FIG 220—Conditions favourable for suction surges

cut off from the motors. It is true that while he is doing so the water may run backwards and drive the pump wrong way round; but no harm will result during this short emergency if the plant has been designed suitably. (*b*) The other reason concerns propeller pumps. *No* sluice-valve or the like can be permitted in a circuit including such a pump, for if it were accidentally left closed the motive unit would be dangerously over-loaded, Fig. 149. This means that other control systems must be evolved adapted to unattended plants and to propeller pumps in particular.

Such a system is shown in Fig. 221 (i); it comprises nothing but an axial-flow pump and a short open pipe. The pump is set at such a level that it is permanently submerged in the suction chamber, and the water is discharged just above the highest surface level on the delivery side. The installation is in every way suited to automatic or to remote control,

§ 281. As soon as the driving motor is switched into circuit, the pump picks up the water without the need for priming; as soon as the motor is switched off, the pump slows down, stops, and possibly runs backwards-way for a few revolutions while the water in the pipe empties itself back into the suction well. Then everything falls quiescent.

Only one serious objection can be brought against this admirably simple arrangement—it is necessarily wasteful at low heads. If the prevailing static head H_{min} is less than the limiting value H_{max}, there can be no reduction in the energy input to the pump; all that happens is that energy is dissipated as the water falls uselessly from the pipe outlet. A very slight

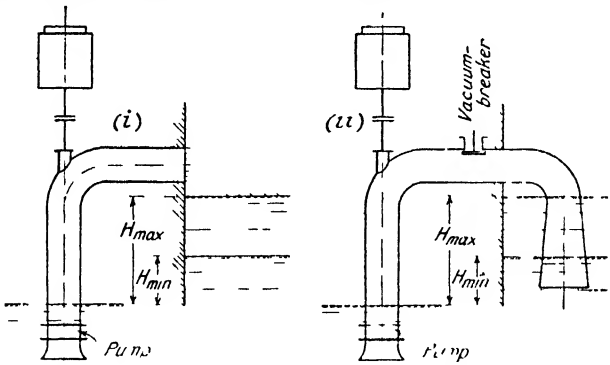


FIG. 221.- Comparative layouts for axial-flow pumps.

modification overcomes this trouble; by submerging the end of the delivery pipe and flaring it outwards, Fig. 221 (ii), highest possible efficiency is ensured at any head. But now another difficulty presents itself: how can we prevent the water siphoning back when power is switched off? The answer is: an automatic vacuum-breaker must be set in the crown of the siphon, diagram (ii). If this is positively opened whenever the pump slows down, atmospheric air will enter, the water will at once drain away, and further flow from delivery side to suction side of the system cannot occur.

(Example 45)

329. Automatic Devices. One type of automatic vacuum-breaker is shown schematically in Fig. 222. The air valve opens vertically downwards, and unless it were restrained it certainly would remain open under the influence of its own weight and

of the pressure difference acting on it. During normal pump operation, a lever and trigger mechanism keeps the valve shut, as the diagram indicates. The elements that respond to speed changes are (i) a small gear-wheel type oil-pump, and (ii) a servo-motor fed from this auxiliary pump. The auxiliary pump can either be driven by its own auxiliary motor, or it can be driven from the main pump shaft through the medium of a centrifugal clutch. In any event matters are so arranged that when the gear-wheel pump runs at a speed corresponding to main pump speed, it generates a pressure in its own circuit high enough to enable the servo-motor to overcome the thrust of the opposing spring. Because the servo-motor has a " leaky " piston, or some equivalent device, oil circulates continuously. Thus the trigger is able to hold the air-valve shut, Fig. 222.

Automatic tripping occurs in this way : as soon as the main pump slows down for any reason, the auxiliary pump stops, the oil pressure fails, the servo-motor piston is pushed back, the trigger releases the lever and the air-valve is free to open. In an electrically-driven set, the auxiliary pump motor is wired in parallel with the main motor so that the two are obliged to start and stop in unison ; in an engine-driven set, the centrifugal clutch throws the gear-wheel pump in and out of service.

It is an automatic system of this type that safeguards the siphonic pumping set illustrated in Fig. 204 (I). As for its use in the conditions described in Fig. 221 (ii), we have to remember that the installation is now no longer fully automatic ; although the air-valve is *tripped* without any manual attention, it must be *set* again by hand.

330. Automatic Gates, etc. Large pumping plants rarely lend themselves to the simplified dispositions seen in Fig. 221 ; there is now no escape from the need for control elements in the pipes, and all that can be done is to see that they cannot create harmful shocks. We might let Fig. 223 (i) represent a typical layout. If it appears that a freely-swinging outlet flap-valve would be unsafe, § 290 (iii), there is a chance that it could be made serviceable by fitting to it the elementary form of dashpot shown in diagram (ii), which exerts an appreciable cushioning effect as the flap-door approaches its seating.

A more elaborate type of hinged outlet door embodies the principle utilised in the automatic atmospheric valve, Fig. 222.

The combination of gear-wheel pump and servo-motor is now adapted to hold the door positively open while the main pump is running, and to lower the door quickly yet harmlessly on to its seating when the pump slows down. Such an arrangement would be very suitable for use with the inclined propeller pump installation, Fig 204 (II); it has the added merit of reducing eddy loss at outlet.

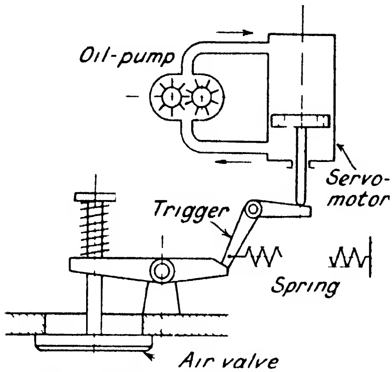


FIG. 222.—Servo motor controlled vacuum breaker.

When the plant necessarily included sluice-gates as in Figs. 202 and 206, these can be made auto-

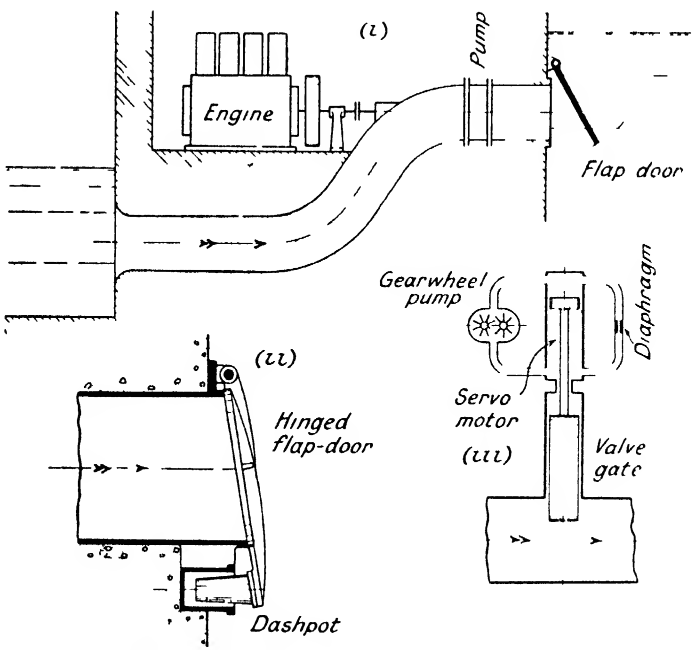


FIG. 223.—Protective devices for low-head systems.

matic by fitting hydraulic operating gear to them (*). The principle can be understood from Fig. 223 (iii). Once again we

rely upon pressure-oil as a working medium, and again, as in § 329, it is fed to the servo-motor cylinder by a gear-wheel pump which responds to speed variations of the main pump. If the system were applied to the half-axial pumping plant described in § 308, automatic operation of the gate would proceed thus : So long as the main pump is at rest, the weight of the sluice-gate will keep it shut. Immediately after the engine has started, the centrifugal clutch engages and the auxiliary pump begins to deliver oil to the operating cylinder ; in consequence the gate steadily opens, and flow from the main pump begins. Before long the set has attained its full speed and discharge, and the gate has reached its highest position, where it is held by the continuous pressure generated by the oil pump. Only when the engine speed falls again is this pressure relaxed, whereupon the gate quietly sinks and the sluice-opening is closed again. The rate of opening and closing can be regulated as desired.

This system can successfully be applied to large installations driven either by oil engines or by electric motors, and including either centrifugal or screw-type pumps.

331. Reduction of Slam-Pressures, (ii) *In High-head Plants.* We now pass to pumping installations where the head is so relatively high that none of the expedients touched upon in §§ 328 to 330 is any longer applicable (*). Henceforth we must rely upon something in the nature of the reflux valves described in § 290. Why will not these standard fittings themselves serve ? They are manufactured and sold in large numbers, so presumably their users are satisfied with them. A partial answer can be found in § 273, which suggests that the severity of slam-pressure is directly influenced by the character of the installation. It follows that while a normal type of reflux valve would fulfil all requirements in one pumping system, the same valve might generate most dangerous pressures in another system. We must remember, too, that we are only likely to depend upon the reflux valve in exceptional conditions, e.g. in the event of the accidental tripping-out of the driving-motor, § 267 ; usually the pumping set is shut down by the carefully-controlled closure of the sluice-valve. So in these circumstances it might be no more than a fair risk to let the reflux valve slam rather hard from time to time ; we should verify

that the pump and piping came to no visible harm, and we should hope that the valve would not slam once too often.

As a criterion for distinguishing between relatively " safe " conditions and potentially harmful ones, one might use a *retardation ratio* having the value $\frac{\text{static head}}{\text{length of pipe}}$. According to § 273, this value represents also the ratio between the retardation of the liquid column at the moment of valve closure, and the acceleration of gravity, g, *assuming that at that instant* the pump itself is generating no head. In a rough way, we can believe that if the retardation ratio is low, there is a chance that a normal reflux valve will serve, while if the ratio is high we ought to look for something else. These tendencies are manifest in Fig. 192, where an increase in H/l from 0·36 to 0·77 corresponds to a greatly augmented shock pressure.

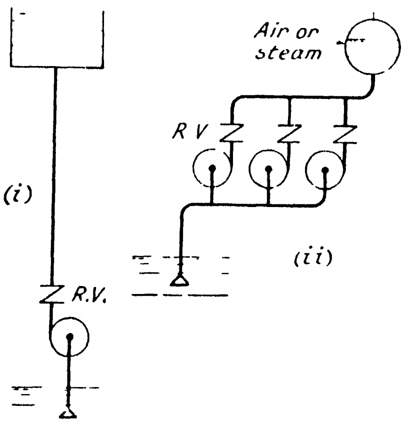


FIG. 224.—Conditions likely to provoke severe slam-pressure.

332. Some Difficult Conditions. Characteristic lay-outs that have awkward possibilities are sketched in Fig. 224. If the pump discharges directly into a vertical pipe (i), the retardation ratio has the value unity—the liquid column is subjected to the full force of gravity. If the pump that is to be shut down is one of a group set in parallel across a pair of bus pipes (ii), the retardation ratio may be much greater than unity. So it may be, also, if there is any kind of air-loaded water container in the circuit, e.g. as in the automatic boosting plant, § 325 ; the compressed air is capable of impressing very rapid retardation on the short water column in the loop circuit. In every way, indeed, the pump reflux valve in this installation is exposed to unusually severe treatment. Because of the automatic working of the plant, the valve gets no protection

from any hand-operated valve, and moreover it may close many times in the course of a day.

No numerical limits can be assigned to "safe" and "unsafe" values of the retardation ratio: the whole conception gives general guidance only. We can only say that the conditions just enumerated are *potentially* dangerous, and we can only depend upon experience for further help.

333. Controlled Reflux Valves. If a standard type of reflux valve appears to be manifestly unsuited for a given situation, how can it be modified to make it suitable? It seems as though two contradictory principles might be applied. According to § 273, the first essential is to ensure that the pivoted valve closes on to its seating the instant forward flow ceases, for only thus can we prevent the incipient return flow that would infallibly bang the valve shut. But with equal plausibility we might argue that return flow would not matter, so long as the valve were mechanically restrained from slamming home. If return flow through a by-pass valve is advantageous, why cavil at reverse flow through the main valve? In fact, both arguments are sound if consistently applied.

(*a*) *Spring-controlled Valve.* Accepting the first of these two principles, we can formulate these conditions: (i) the lift of the valve must be restricted; (ii) the inertia and thus the weight of the moving parts must be small, (iii) there must be a powerful closing force. The instant the forward flow of the liquid slackens as the pump slows down, we want to give the valve-flap every encouragement to hasten back on to its seating so that it is safely home as the last drop of liquid squeezes past. By using an external spring to supply the closing force, we arrive at some such construction as is sketched in Fig. 225; but we also impose a serious energy loss on the liquid as it forces itself through the restricted opening during normal pump operation. In electrically-driven pumping sets the difficulty can be overcome by using a solenoid-operated trigger to hold the valve wide open while the pump is running (*). As soon as the supply current fails and the solenoid is no longer energised, the trigger is released, and the main spring takes charge. Naturally the apparatus must be re-set every time the pump is started, Fig. 225.

(*b*) *Dashpot-controlled Valve.* The simplest embodiment of

dashpot control was illustrated in Fig. 223 (ii), where a projection on the swinging flap served as the dashpot piston. As applied to a reflux valve for use in a closed pipe, this principle of damped closure would require a more complex arrangement of levers, links, and an external dashpot cylinder. These would be adjusted so as to lower the valve gently on to its seating in spite of the return flow which was trying to slam the valve.

In effect, there may be various combinations of systems (*a*) and (*b*); moreover, exceptionally skilled design of freely-pivoted valves may enable them to close quietly even in unpromising conditions (*).

334. Reducing Negative Surge. As already explained, slam-pressures have the nature of a sudden, violent "kick"; but the true surges now to be re-examined develop relatively slowly and the wave form can be predicted fairly accurately. The time scales of the autographic records reproduced in Figs. 191 to 193 are significant; whereas for true surges the graduations are spaced apart at intervals of 5 or 30 seconds, they must represent only one-tenth of a second when slam oscillations are to be shown.

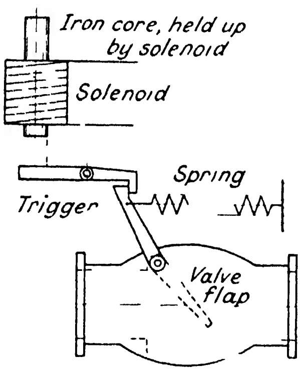


FIG. 225.—Rapid-closing reflux-valve.

How can the intensity of the negative surge in the delivery main be controlled? The *length* of the pipe—a matter of perhaps 500 or 5000 yards—will probably be fixed by the basic needs of the installation. The velocity of the *pressure-wave* along the pipe, § 261, is virtually unalterable. The pipe *diameter* might be increased so as to lessen the water *velocity* at the moment of tripping out the pump motor, but the cost of the pipe would rise prohibitively. There remains the possibility of altering the *rate of retardation* of the rotating parts of the pumping-set, § 267. It is a very attractive possibility, too. Both Fig. 189 and Fig. 190 reveal a clear connection between this rate and the intensity of negative surge: the moro gradually the pump rotor slows down, the smaller is the surge.

(*a*) *Use of Flywheel.*

Suppose, then, that after plotting as in Fig. 189 (iii) the wave form for a particular installation, we find that it falls below the pipe ? The negative head thus disclosed could not be tolerated, especially if it were great enough to encourage separation, § 264. The intensity of surge must be reduced ; the rotating parts must be encouraged to keep up their speed ; and this can be done by *increasing their mass*. There is no technical difficulty in doing this. The half-couplings on motor shaft and pump shaft may be made specially heavy, or a flywheel may be mounted on the motor shaft. This would only mean that the main bearings might need a little modification to fit them to take the additional weight, and that possibly the motor starting gear should be re-designed. In practice the system is completely successful, and is regarded as the most effective of all methods of surge-pressure reduction (*).

(Example 46)

(*b*) *Use of Air-chamber.*

An alternative—and sometimes complementary — method depends upon a large *air-chamber* communicating with the lower end of the main delivery pipe, at a point as close to the pump as possible. If the pressure in the pipe tends to fall, consequent upon a drop in pump speed, the compressed air in the chamber forces out some of the stored water and helps to maintain flow in the pipe, Fig. 226.

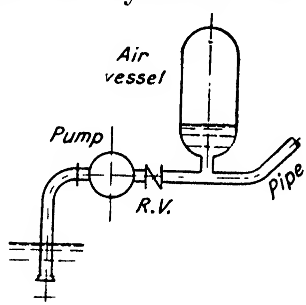


FIG. 226.—Use of air-vessel for controlling surge-pressure.

335. Reducing Positive Surge. The two principles to be applied here are : (i) Do all that is possible to keep down the intensity of the negative surge. (ii) If this is not sufficient, provide an *automatic relief-valve*, Fig. 193, § 274. Various types of relief-valve are available ; in some of them, the impulse which initiates the opening movement is provided by the pressure drop in the pipe itself, while in others there is a servo-motor mechanism which responds to changes in pump speed, as in § 330.

The *air-chamber* mentioned above is not as efficacious in cushioning positive surges as might be imagined, for the

secondary surges generated by the expansion of the air may themselves be troublesome (*). Moreover, the large volume of compressed air so close to the pump may itself encourage serious *slam*-pressure when the main reflux valve closes, § 332.

336. Special Methods. In the largest size of medium-head or high-head pumping installation, the energy of the water column in the delivery pipe, at the moment of tripping out of the main pump motors, is so formidable that only the most positive measures of control are likely to promise security. In consequence, any reasonable expenditure on protective

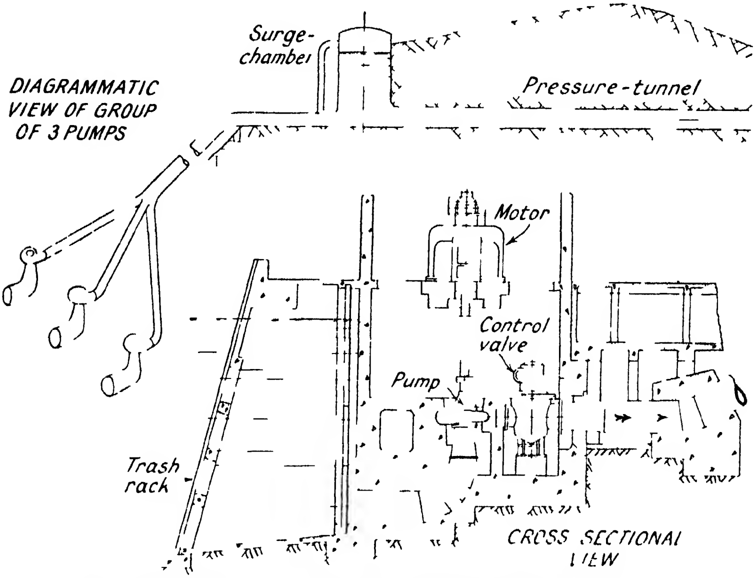


FIG 227 —Pumping plant for the Colorado River aqueduct

devices can be regarded as no more than a justifiably incurred insurance premium. As an example of the most advanced practice in this field one may cite the mechanism designed for the pumping installations of the Colorado River Aqueduct (U S A); these plants repay study all the more because they illustrate so many of the principles explained elsewhere in this book.

The various plants are built up of groups of pumps arranged as shown in Fig. 227. Each group has three vertical electrically-driven units, and each pump has its individual delivery pipe (*).

The three delivery pipes unite to form a single large rising main which terminates in a capacious surge-shaft; thence the water flows through a pressure tunnel to a balancing reservoir where it is picked up by further pumps and passed on through other open or closed conduits. Some general points to be noticed are :—

(i) Being of the side-inlet, medium specific-speed, centrifugal type, and of exceptional size, the constant-speed pumps work at an efficiency in the region of 93 per cent.

(ii) This efficiency can consistently be maintained in service, because the balancing reservoir and the disposition of the pumps in parallel permit the start-and stop system of flow regulation to be used, § 241 (ii).

(iii) Cavitation risks are minimised by running the pumps under a positive suction pressure, or *negative* static suction lift, § 255 (i). The pumps are set *below* the river surface level.

(iv) The chain of stations spaced along the aqueduct represents a crowning example of the multiple-station system, § 322, fed with energy by its special electrical distribution lines, § 299.

337. Power-operated Control-Valve. A single main valve interposed in the diverging pump-outlet pipe, Fig. 227,

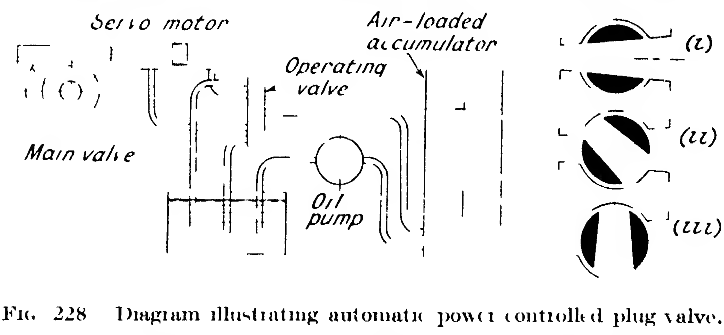


FIG. 228 Diagram illustrating automatic power controlled plug valve.

is designed to serve both as an isolating valve and as an automatic emergency relief valve. In essence this organ is a very large plug-cock, specially built so that during the turning operation the plug is raised slightly in its housing, so as to reduce the energy expended in this process. The source of energy is an electrically-driven oil pump, Fig. 228. It feeds pressure-oil

into an air-loaded accumulator or reservoir; thence the oil passes through an operating-valve to the main servo-motor which does the actual work of opening or closing the main plug-valve.

During routine working of an individual pumping set the valve is used in the normal manner, § 266. It is left closed while the pump is being run up to speed, and then, by regulation of the operating-valve, the servo-motor is caused to open the main valve at the necessary slow rate. The valve remains fully open during normal running, Fig. 228 (i). When a pump is to be taken off load, the plug-valve is slowly closed and finally the pump motor is switched off.

It is during an emergency shut-down, following a failure of the electric power supply to the station, that the air chamber proves its value; it serves as an alternative source of power for energising the control mechanism, even though the oil pump has fallen out of action. Immediately the electric elements go dead, an automatic device takes command of the servo-motor operating-valve and causes the main valve-plug to turn rapidly nearly into the closed position, Fig. 228 (ii). Meantime a negative pressure-surge has inevitably been developing in the rising main. When the ensuing positive surge arrives at the main valve, it finds an opening - albeit a steadily diminishing one—through which it can dissipate itself; return flow takes place back into the pump rather than through a separate relief valve as in Fig. 193. But such return flow can do no harm, and it is gradually extinguished as the plug-valve continues to turn slowly to its final fully-closed position, Fig. 228 (iii).

There is an instructive comparison to be made between this very elaborate system of surge-pressure reduction, and the simpler automatic device illustrated in Fig. 223 (iii). In pumps specially built for hydraulic-storage plants, still another method of surge control can be embodied, § 350.

Special Priming Systems

338. Some Remaining Possibilities. In summing-up the problem of priming as it affects the whole installation, § 291, the various types of solution can be rearranged thus:

(i) If the pump is permanently set below the level of the

liquid on the suction side *no* priming equipment is needed at all, Figs. 204 (II), 207, 219, 221, 227.

(ii) If the pumps are of medium or large capacity, working normally against a suction lift, Figs. 199, 202, 205, there will be independent evacuating pumps or ejectors that are set in operation by the attendant during the routine of starting a main pump, § 292.

(iii) Self-priming pumps of special construction embody priming apparatus that requires no manual attention, §§ 152, 156.

There remain to be described a few devices and dispositions that do not exactly fall into any of these categories, and that offer particular solutions adapted to special conditions.

339. Systems for Small Engine-driven Pumping Sets. The engine itself or its accessories can help to prime the pump it drives, thus :--

(i) After the engine has been started, the exhaust can be partially throttled so as to create an appreciable back-pressure ; and as the pump is not yet under load, this temporary throttling does not incommode the engine (*). A part of the exhaust gas is then diverted through the nozzle of an ejector (of the type shown in Fig. 195) built into the pump, and priming proceeds normally. The exhaust is freed as soon as the pump picks up its water. The whole apparatus, which is intended only for high-speed engines of a few horse-power, can be thrown in and out of action by the movement of a lever.

(ii) The partial vacuum created in the induction system of the engine can likewise be used to draw air out of the pump and suction pipe. Assuming that a more or less standard type of automobile engine is in question, a small-bore pipe could be connected to the induction manifold and to the centrifugal pump casing ; a float-valve interposed in this pipe, § 292, would safeguard the engine from flooding. It is just during the period of priming, when the engine is running nearly light, that it develops the highest vacuum —a fortunate coincidence.

(iii) Another accessory that can be pressed into service in these small pumping-sets is a diaphragm-type petrol-pump, of the sort used as a fuel-pump in a motor-car. It is now made to work as an evacuating pump. A special unit is employed, mounted in such a manner on a hand-lever that a

movement of the lever brings the actuating trigger of the evacuating pump into engagement with a cam in the engine. Priming of the main pump then begins, the air being drawn out by a flexible pipe coupled to the suction side of the auxiliary pump. The hand-lever is thrown over again as soon as the main pump begins to work normally (*).

(iv) If a rotary type exhausting pump is preferred, § 292, it can be mounted on a rocking frame pivoted about an axis parallel with the main pump axis. This auxiliary pump is driven through friction wheels keyed on the main pump shaft or the engine shaft, and its inlet passage communicates with the main pump casing by a ported trunnion. When the priming pump is in the " off " position, the port is closed and the pump is idle; when the engine has been run up to speed and the priming pump has been swung over into the " on " position, the friction wheels engage, the auxiliary pump begins to work, and priming begins.

340. Automatic Priming Installations. While it is not pretended that a sharp distinction is here maintained between a self-priming pump and an automatic priming installation, yet such a distinction could in principle be recognised. A self-priming *pump* is a specially-built machine; an automatic priming *installation* comprises a standard centrifugal pump to which has been added appliances which permit priming to proceed without manual control. The diverse dispositions available might be separated into the categories of *electric* and *hydraulic*.

(i) *Electric Systems.* Here the essential new component is a set of electric contacts or the like which automatically energises some standard evacuating appliance whenever the main pump needs priming (*). Two systems are shown schematically in Fig. 229. In scheme (*a*), an air chamber connected to the main suction pipe accommodates a float which is adapted to operate " start " and " stop " contacts. When the float falls to its lowest position, e.g. when the suction pipe is dry, the electrically-driven vacuum pump will be switched on and air will be drawn out of the suction pipe and pump casing. The vacuum pump will be switched off as soon as the water level in the air chamber has risen to its upper limit. In scheme (*b*), the priming apparatus is mounted above the main pump casing, and the

electric circuit is controlled by electrodes. If the water level in the chamber drops too low, a solenoid-operated valve admits pressure-water to the nozzle of an ejector, and the excess air is removed.

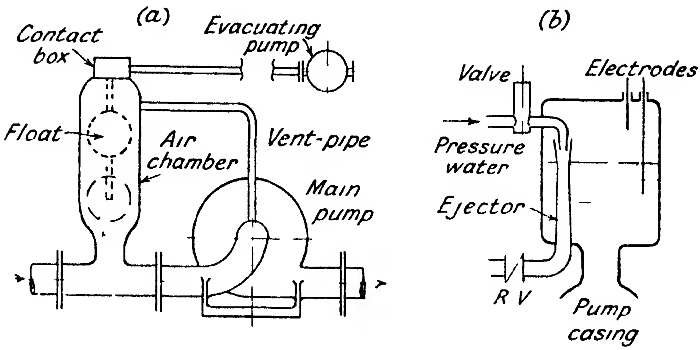


FIG 229 –Electric automatic priming systems.

(ii) *Hydraulic Systems.* These are shown in Fig. 230. The "Seaborne interceptor", (*a*), consists of a cylindrical drum, fixed in the pump suction pipe, whose capacity exceeds that of the whole suction system, a loop in the pipe ensures that after the pump is stopped, the pump and the drum remain full of

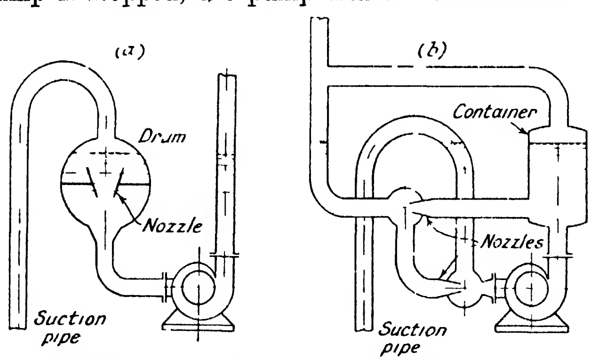


FIG 230 Hydraulic automatic priming systems.

water. The priming element is a conical nozzle housed in a horizontal diaphragm extending across the drum. When the pump is started, water flows into it from the drum, and more water rises up the suction pipe to replace what has left the drum. Before the reserve supply in the drum is exhausted, the suction pipe is fully primed and "solid" water has begun to enter the

top of the drum. In passing through the nozzle, this water is able to entrain in it the residual air in the drum, and the mixture of air and water passes through the pump. Manifestly the lay-out of the suction system will here substantially reduce the permissible suction lift on the pump.

Scheme (*b*), Fig 130, has points of similarity with the jet type of self-priming pump, Fig. 102 ; but now the priming jet does its work before the water enters the pump (*). The object of having two nozzles is this : during the priming period, when no water flows up the delivery pipe, low-pressure water passes in turn through the two nozzles, but when the pump is running normally the two nozzles work in opposition and thus discourage return flow from delivery to suction side of the system.

Auxiliary Pumping Plant in Steam Power Stations.

341. Categories of Pump. The auxiliary pumps installed in a large steam-turbine power plant are not in principle of unusual design (*). What gives them as a whole their peculiar interest is the variety—and in some cases the highly-specialised nature—of the services they perform ; and as these services are highly responsible ones, involving the continuity of operation of the main units, the pumps must above all else be reliable. Another point is this : in an electric generating station, both steam and electricity are available on exceptionally favourable terms, and there are thus unique opportunities of studying the rival merits as motive units of auxiliary steam turbines and electric motors.

The three types of pump chiefly needed are :—

(i) Low-head pumps of large capacity for forcing condensing water through the condenser tubes and the circulating-water system.

(ii) Condensate-extraction pumps for drawing condensed steam from the condensers against a high vacuum.

(iii) Feed-pumps for feeding into the boilers the water received from the extraction pumps.

There may also be various other pumps of less importance that cannot individually be mentioned here.

342. Circulating-water Pumps. (i) *Supply taken from River or Cooling Pond.* The hydraulic system to which the

circulating-water pumps must conform can generally be represented by some such diagram as Fig. 231. Water from the river flows in turn through the pumps and the condenser, and returns again to the river under siphonic conditions (*). Thus the head, h_m, on the pumps, broadly speaking, is wholly a frictional one; it comprises the energy loss in the condenser passages and tubes, which may be of the order of 15 ft. head, and the losses in the inlet and outlet conduits which may be hundreds of yards long. But when starting up the system, the outlet leg of the siphon may not be charged with water, and then the pumping head is considerably greater than normal. Moreover,

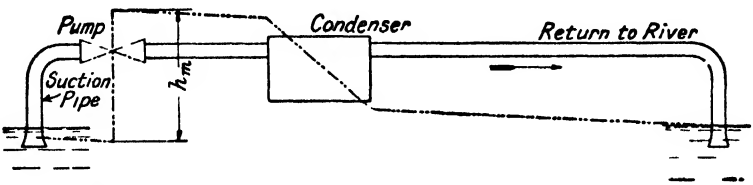


FIG. 231.—Hydraulic gradient in circulating-water system.

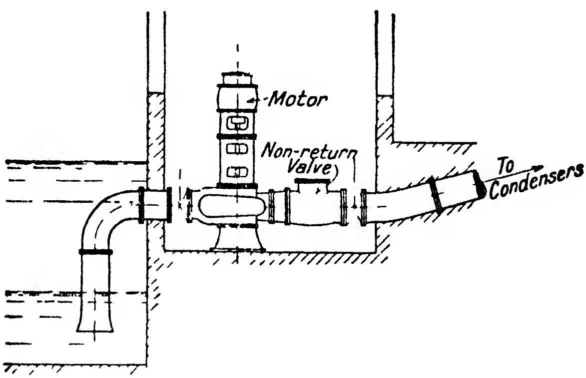


FIG. 232.—Vertical-shaft circulating-water pump.

as the source of water may be a tidal river or estuary, or at least a channel subject to fluctuations of water level, the suction lift on the pumps will vary very widely.

As a rule, the accommodation available for the pumps is either in the basement of the main power house or in a special pump-room built on the bank of the river. Such an

installation of pumping-sets housed in their own building is illustrated in Fig. 232.

To meet these varied conditions, standard horizontal double-entry centrifugal pumps of medium to high specific speed are usually preferred. Sometimes the twin-rotor type is chosen, § 116, or the twin-pump disposition, § 320 (*a*). Vertical pumps fit in very well with the arrangement of Fig. 232. The advantage of axial-flow pumps is that the high head generated at low discharges, § 213, enables them to "throw" the water up to the condensers before siphonic flow has been established, provided that the motive units have a sufficient overload capacity to withstand these abnormal duties. **(Example 47)**

Electric motors are invariably used as motive units. Although constant-speed units are often considered satisfactory, the hydraulic conditions are eminently suited to variable-speed control of discharge (*****), imposing as they do a purely frictional head, § 242 (ii). Certainly the overall efficiency of the whole plant can be improved if the rate of flow through the condensers can be adapted to the temperature of the river, the rate of steaming of the boilers, etc. Scherbius speed control of centrifugal pump motors has been used, § 278, and variable-slip hydraulic couplings for axial-flow pumps, § 284.

(ii) *Cooling-tower Stations.* Here the head and the suction conditions are virtually constant. While centrifugal pumps are commonly specified, there is an opportunity also for constant-speed two-stage variable-pitch propeller pumps, § 127; they give very economical discharge regulation.

(iii) *Marine Installations.* For use on shipboard, vertical-shaft motor-driven centrifugal pumps have the particular merit of compactness. In high-speed naval light craft, preference is often given to propeller-pumps each driven by a steam-turbine through a step-down gear-box.

343. Feed-water Circuits. In Fig. 110, § 163, there was presented in its simplest form the problem of transferring water from condenser to boiler. That diagram now has to be elaborated and corrected so as to agree more closely with what really happens to the water during its journey, thus :—

(i) Two or more pumps working in series must share the work between them, viz., an extraction pump and one or more feed-pumps (*****).

(ii) The energy loss in the system no longer depends solely upon pipe friction, as was shown in Fig. 110 ; in addition there are pressure losses imposed by feed heaters, regulators and the like, interpolated in the circuit.

(iii) As the quantity of water pumped conforms to the variable rate of steaming of the boiler, so will the head generated by the pumps vary in accordance with their particular head-discharge characteristics.

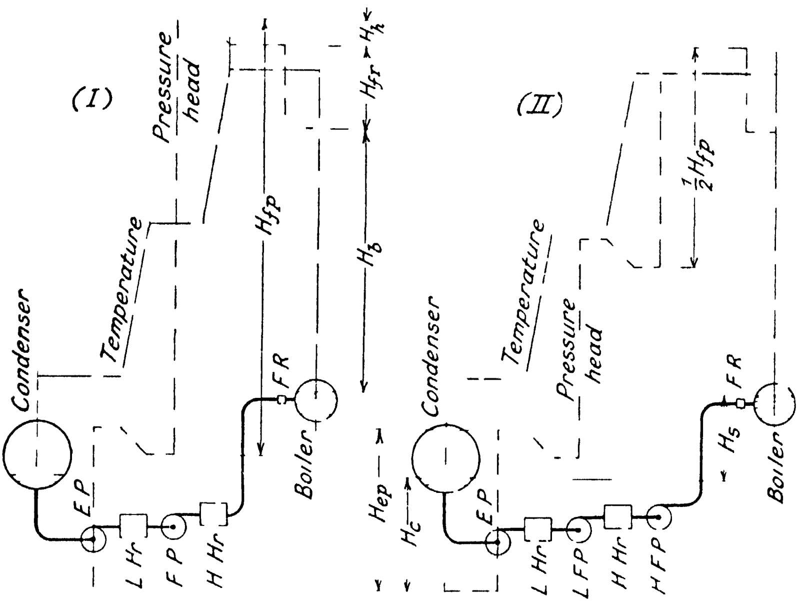


FIG. 233 — Feed water circuits (*E.P* — Extraction Pump, *F P.* – Feed Pump, *Hr* Heater, *F R* Feed Regulator)

(iv) Most important of all, the water during its transit receives not only *hydraulic* energy from the pumps but it also receives *heat* energy from the heaters. The resulting change in the physical properties of the water reacts upon the pump performance in two ways : (i) the rise in vapour pressure affects the suction capabilities, § 253 ; (ii) the fall in density increases the energy input to the pump per unit *weight* of liquid, § 230.

In order to include only essential information in our new diagram, we may consolidate the hydraulic losses, (ii) above, into three main items associated respectively with (*a*) a low-temperature heater or group of heaters, (*b*) a high-temperature

heater or group of heaters, (*c*) the automatic feed regulator which controls the rate of flow into the boiler. By this means, and by plotting *temperature changes* as well as pressure changes, we arrive at something in the shape of Fig. 233 (I). This circuit is manifestly hard on the high-temperature heaters : it subjects them to a pressure which is substantially higher than full boiler pressure. An alternative system is to divide the work between two feed pumps, with the high-temperature heater set between them, Fig. 233 (II) ; but this arrangement protects the heaters at the expense of the high-pressure pump, § 345.

When the type of circuit has finally been settled, the business of choosing the individual pumps can proceed.

344. Condensate Extraction Pumps. Of the general construction described in § 145, these machines may have one or two stages, and they can be set horizontally or vertically. In land installations they are almost invariably driven by constant-speed electric motor, while on shipboard they may be turbine-driven. The sketch reproduced in Fig. 234 explains why such pumps must be able to draw against the highest possible vacuum. The water collected at the bottom of the condenser shell, waiting to be evacuated by the pump, is itself under saturation conditions : it would boil if the absolute pressure on it were in the slightest degree reduced. But we have already agreed that nowhere in the pump can there exist a pressure *lower* than this, § 247. If, therefore, the pump is now just on the point of cavitation, it follows that the positive head h_{sp} is all we have to rely upon for overcoming *all* relevant changes of pressure-head, viz., the items H_{ext} and H_{int} specified in § 252.

The value of this positive head—which is represented by the vertical distance between the water level in the condenser and the axis of the extraction pump—has a direct bearing on the layout of the whole plant. In a land installation the advantage of keeping the distance as short as possible is that the extraction pumps can be accommodated in the basement below the condensers without the need for excessive excavation ; in a marine installation there is the question of ensuring a low centre of gravity for the condensers (*).

Naturally the rules of § 253 are not applicable here, because

the pump is certainly not of standard construction. In regard to the variables listed in § 252, none of those numbered (i) to (vi) has now any influence; only those directly expressible in *head of liquid* need to be taken into account. Even then, the values of the individual items (vii) to (xi) can rarely be estimated with sufficient accuracy to permit the necessary positive head h_{sp} to be assessed; only the pump maker can provide this information. It may be as low as 2 ft.

345. Boiler-feed Pumps. (i) *Type.* For service on land, horizontal multi-stage centrifugal pumps are commonly installed, either of the ring type or the barrel type, § 121. At sea, the compactness of the direct-driven turbine-operated single-stage pump is a recommendation (*).

(ii) *Motive Unit.* When the pump forms part of an electric generating plant, the most economical form of drive will undoubtedly be by direct-coupled electric motor. Constant-speed sets are frequently accepted in spite of the energy loss in the feed regulator, Fig. 233. If the maximum attainable motor speed will not give the desired pressure without an inordinate number of pump stages, a step-up gear-box might be interposed (*). Variable speed direct-coupled sets are by no means unknown.

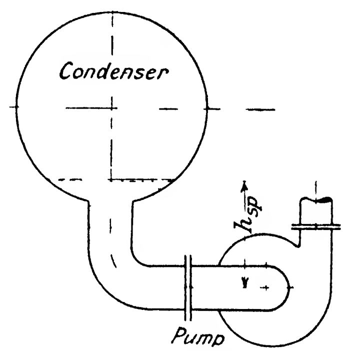


FIG 234 —Location of extraction pump.

For emergency and stand-by use there must also be a turbine-driven pump, that can take up the load at once—preferably automatically—if for any reason electrical energy is cut off from the pump motors. As just mentioned, all the feed-pumps in a marine engine-room are likely to be turbo-driven.

(iii) *Construction.* Various modifications in design may be needed to fit the pump for continuously delivering hot water; these have been explained in §§ 138 to 144. The most severe conditions will arise—a combination of maximum pressure and maximum temperature—in the high-pressure pump of the group

shown in Fig. 233 (II). It will be the stuffing-box on the inlet side that will probably need special care, § 143.

(iv) *Estimation of Head and Power Input.* The hydraulic gradients plotted in Fig. 233 give guidance in estimating the effective head on the pumps. Using the symbols : H_s = static head, H_c = head equivalent to condenser vacuum, H_b = head equivalent to boiler pressure, nH_h = gross losses in heaters, etc., H_{fr} = loss in feed regulator, H_{ep} = effective head generated by extraction pump, H_{fp} = effective head generated by feed pump (s) ; then

$$H_{ep} + H_{fp} = H_s + H_c + H_b + nH_h + H_{fr}.$$

The water passing through the pump will almost certainly be hot enough to make the corrections detailed in § 230 essential. For a stated hourly *weight* of water delivered to the boiler, the feed pumps will absorb least energy if they handle the water at the *lowest* temperature. On these grounds, system (I) in Fig. 233 is more economical than system (II).

(v) *Control of " Suction " Conditions.* The term " suction " can here only be used in a conventional sense, for when the water enters the feed pump its pressure will be above—probably far above—atmospheric, and not below it. Nevertheless the question of pressure conditions within the impellers is no less important than it was judged to be in § 252. Because the exceptional conditions foreseen in § 255 (ii) are now operative, the procedure recommended in § 253 should be put aside in favour of the following :—

(*a*) From the known water temperature at inlet to the pump, the vapour pressure or saturation pressure can be found, p_{vp}.

(*b*) From equation (16-3), § 252, the internal head drop H_{int} can be computed. The effective head H_e will be *that of the first-stage impeller,* and as this impeller will be of more or less standard design the appropriate value of the cavitation factor σ can be read from Fig. 180.

(*c*) If w_{if} is the water density at the inlet flange, then the *minimum absolute pressure at the inlet flange* will be represented by : $p_{vp} + w_{if}H_{int}$. Naturally it will be prudent to allow a margin of safety of 10 or 20 lb./sq. in. above this value. A truly-plotted pressure diagram on the lines of Fig. 233 will show whether this margin exists or not. In other respects,

too, graphs such as these may be more immediately useful if they are expressed in terms of pressure rather than of head.

(Example 48)

346. Feed-pumps: Other Questions. (i) *Characteristics of Pumps and of Circuit.* The general relationship between discharge and head, for constant-speed pumps in series, has already been plotted in Fig. 168. In that diagram we may take pump A to represent the extraction pump, pump B to represent the feed pump, the head H_s to represent the total (constant) *effective* head difference between the water in the condenser and the water in the boiler, and the head H_f to represent the gross hydraulic energy loss in the circuit at maximum pump output. At reduced rates of flow the vertical distance between the external curve and the internal combined characteristic curve ($A + B$) will denote the additional head to be dissipated in the feed regulator, § 343. Of course Fig. 168 makes no pretence at showing these various quantities truly to scale. In fact the external head line will not depart greatly from the horizontal, which suggests to us that the feed regulator will waste least energy if the combined pump characteristic is also as flat as possible, § 240 (i).

(ii) *Parallel Working of Pumps.* Now flat characteristics are almost necessarily of the *unstable* type, § 215, and these are inadmissible if, as frequently happens, two pumps are discharging in parallel into a common feed range. Pumps with such characteristics might not share the water equally between them. Instead, first one pump and then the other would try to get the bigger share, the final result being a violent and even dangerous periodic surging. A steeply-falling characteristic would certainly remove this inconvenience, but at the cost of heavy energy loss at light loads. To obtain the desired stability without losing too much energy in the feed regulator is a matter of unusually skilful design.

(iii) *Disposal of Surplus Heat.* When the feed regulator is severely throttling back the flow, the power input to the pump is relatively much greater than the power output, § 206. The difference between the two is wholly converted into heat. As the flow through the pump is itself small, this heat cannot be carried away quickly enough, and in consequence the whole pump and the water in it may become excessively hot. If

these temperature conditions appear to be dangerous, some kind of by-pass or leak-off should be contrived, for leading a small return flow from the delivery side of the pump to a part of the system where it can safely be accepted (*).

Transportable Pumping Plants

347. Mobile Pumping Outfits. Self-contained pumping sets that can be moved from place to place have an immense range of utility. To be truly independent of their surroundings the pumps should manifestly be driven by internal-combustion engines, but occasions arise when electric motors can be used, taking their power through flexible leads. Neither the pump nor the motive unit need be of fundamentally special design; what they both need above everything is the simplicity and strength by which alone they can hope to withstand long periods of rough outdoor service.

General-purpose sets are intended for unwatering foundations, draining pits and sumps, miscellaneous contractors' jobs, and any temporary or emergency work they can manage. Engine and pump are mounted on a wheeled trolley, with fuel tank and accessories, and they may be given some light protection from the weather. The pump must be able to pass mud and sand without complaint; and as it will probably be expected to draw through a suction hose dropped direct into the water, self-priming characteristics will be most desirable.

Fire-fighting pumps have perforce been greatly developed during recent years (*). A common type of set comprises a 2-stage centrifugal pump direct-coupled to a petrol engine, the whole being carried either on a trailer or on a self-contained motor vehicle. It is manifestly quite essential that the pump can be got into action immediately it arrives at the fire: to this end, one or other of the automatic priming systems described in § 339 may be effective.

Salvage pumps have exceptionally rough kind of work to do on wrecks, where they are exposed to salt water both inside and outside. The set must have lifting slings so that it can be lowered into place by a crane.

For use in *oil-fields* and in similar circumstances, pumping-sets may be sufficiently mobile if they are mounted on stiff steel *skids*.

348. Floating Pumping Stations. The type of pump installed for fire-fighting in a *fire-float* will not differ greatly from what is used on land : but it will probably be more powerful.

A *suction-dredger* is a highly-specialised floating pumping plant (*). The dredge-pump may be called upon to handle a mixture of water, mud, sand, and broken rock having a density up to 1·5 : its 4-bladed impeller may be as much as 8 ft.

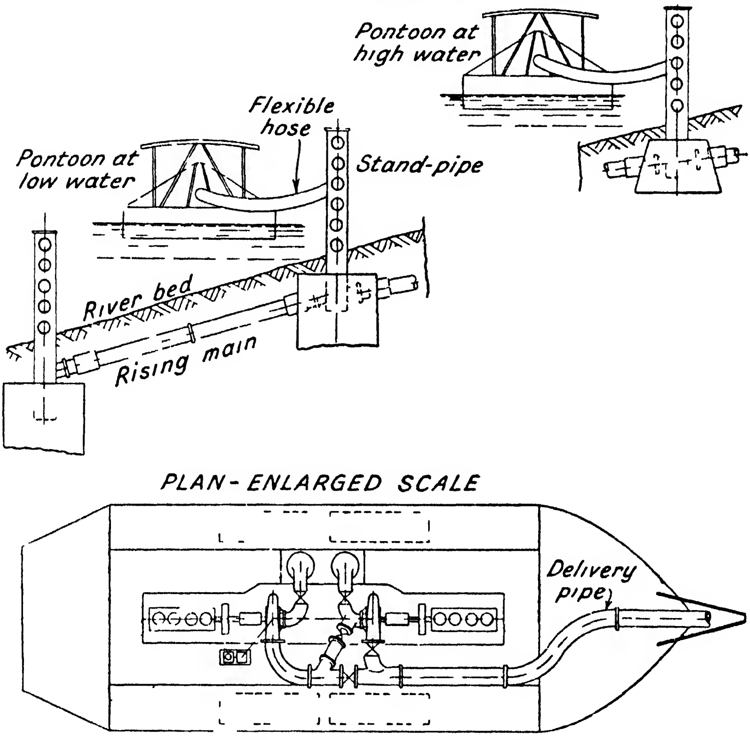


FIG. 235.—Floating pumping plant.

diameter (*) driven by a steam-engine of 2400 h.p. Unusually resistant construction and material are indispensable, § 147.

Lift *irrigation* is sometimes facilitated by mounting the pumping plant on a pontoon. If, for instance, the conditions shown in Fig. 205 were still more accentuated—if the range of river water level were greater, and the slope of the foreshore were more gradual—then at low level it would be almost impracticable to reach the water at all by any kind of fixed

installation. The diagram, Fig. 235, gives an impression of how a floating plant would deal with such a situation. On the pontoon are two engine-driven pumps that can be connected either in series or in parallel, §§ 243, 319; erected in the river bed are vertical stand-pipes linked by a rising-main. There are flanged connections on the stand-pipes for receiving the coupling of the flexible hose which conveys the water from the pumps; the pontoon would be moored close to whichever stand-pipe was rendered most convenient by the state of the river.

Similar floating plants have been used for lowering the level of *lakes*. In such cases the steel delivery pipe might have ball-and-socket joints, and would itself be carried to the shore on a series of smaller pontoons or buoys.

Hydraulic Storage Plants

349. Conditions of Service. A hydraulic storage plant is usually a self-contained system that works in association with a distant electric generating station. When the demand on the main thermal or hydro-electric station is low, surplus energy can be transmitted to the storage plant and there held in reserve. At times of peak load on the main station, the stored energy is released and is used to augment the output of the main station (*).

The storage plant must consequently embody the following elements: (i) a lower collecting basin or reservoir; (ii) an upper collecting basin; (iii) a pump designed to lift water from the lower to the upper basin during storage periods; (iv) a hydraulic turbine which will transform hydraulic energy as the water flows back during generating periods; (v) a rotary electrical unit that can act either as motor or as generator, depending upon whether it is coupled to the pump or to the turbine.

The pumps—for in a given installation we shall expect more than one machine—operate on the intermittent system described in § 241 (ii), the only difference being that here the water afterwards flows backwards from the reservoir instead of flowing forwards. Constant rotational speed is obligatory, for none other than synchronous electrical units are practicable. Provided, therefore, that the range in level in the storage basins is not excessive, all the conditions are present for the utmost efficiency of working. They need to be, too, for the whole

system would have no economic justification if transmission and conversion losses exceeded reasonable limits.

350. Types of Storage Pump. Either single-stage or two-stage centrifugal pumps may be suitable, set either horizontally or vertically. Sectional views of a large horizontal unit were reproduced in Fig. 106, § 157, and the manner in which such a pump is assembled into a complete machine set is shown in the accompanying Fig. 236. During the pumping or storing part of the cycle, the pump is clutched to the motor by a hydro-mechanical coupling while the hydraulic turbine revolves idly in air ; during the generating period, the turbine is under load and the coupling is disengaged so that the pump is at rest (*).

As each pump may deliver something of the order of 5 or 10 tons per second against a head of perhaps 500 or 1000 ft.,

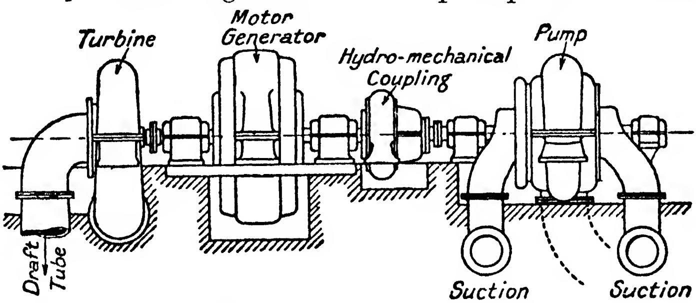


FIG. 236.—Hydraulic-storage plant.

the machine can be built on a scale giving the promise of maximum efficiency, § 225 ; yet the problem of inertia shocks in the long delivery pipe becomes of corresponding gravity. One solution of the problem has been described in § 337. Another solution—the one preferred on the continent of Europe —was represented in Fig. 105, § 157. Instead of using a single control-organ, e.g. a valve, to prevent return flow when the pump stops, a number of pivoted guide-blades are disposed around the pump casing. As soon as power is cut off from the driving motor, either intentionally or accidentally, a suitable servo-motor operates the link mechanism seen in the diagram and quickly shuts the blades. In this manner, slam-pressures are wholly prevented. As for the negative and positive true surges, their intensity is relatively low and calls for no special counter-measures because of the low velocity in the delivery pipes.

CHAPTER XXI

ERECTION, OPERATION AND MAINTENANCE

351. The Pump at Work. It is not without significance that the concluding chapter of Part D of the book should be the shortest. If there seems so little to be said, relatively speaking, about running and maintaining the pump (*), that is surely a tribute to the simplicity and reliability of roto-dynamic pumps in general. In the best conditions, this reliability is so unfailing that a pumping-set may be able to run continuously, non-stop, for two years or more at a time.

352. Erection on Site. Some types of pumping set, e.g. those mounted on trolleys, § 347, require no erection whatever ; at the other extreme are large low-head sets that must be permanently built into the foundations of the pump-house, § 312. Here we may only consider a typical medium-sized direct-coupled electrically-driven outfit, mounted on a cast-iron or fabricated bedplate. Points to be noted are :—

(i) Although when received on site the pump and motor may already be fitted on the bedplate, it is prudent to regard this assembly as provisional only ; the locating pins for the feet of pump frame and motor frame should not be inserted for both units, but for one only. The final drilling and pinning should be left until the bedplate has been bolted down.

(ii) Before levelling up the bedplate on its foundation, both pump and motor should preferably be taken off.

(iii) With the bedplate supported on iron wedges and accurately levelled, and with the foundation bolts grouted in and tightened, the pump and motor can be replaced. The two shafts must be checked for alignment and parallelism in the usual manner, viz., by a straight-edge laid across the half-

couplings, and distance-pieces inserted between them. For a multi-stage pump with balance-disc, the specified axial clearance between the two half-couplings should prevail when the pump shaft is hard up towards the suction end of the pump. When all is satisfactory, the remaining steady pins can be put in and the bedplate finally grouted up.

(iv) After the grout has set, and the suction and delivery pipes have been bolted up, and the nuts of the foundation bolts finally pulled home, the free running of the two shafts should again be verified.

Vertical installations, e.g. borehole pumps, may require special care in assembling. To ensure that the main axis hangs truly plumb, the joint face between the base of the motor and the top of the water collecting box, § 313, may be of dished or spherical form instead of being flat.

If hot fluids are to pass through the set, the question of vertical expansion must be studied, § 139. If, for example, a cold-water pump has a steam-turbine as motive unit, the turbine shaft should deliberately be set a trifle *low*, so that under working conditions the axis will exactly have risen into line with the pump shaft.

353. Preparations for Running. Before the set can be put under load, points to be verified are :—

(i) See that nothing has been left inside pump or piping, e.g. packing material, tools, etc.

(ii) Flush out the oil-wells of ring-lubricated bearings, and fill with a suitable grade of lubricating oil.

(iii) Pack the main stuffing-boxes. Except in special circumstances, §§ 141 to 144, soft cotton packing is best, § 86, impregnated with grease or graphite. The rings should break joint, and the gland-nuts should only be pulled up finger-tight.

(iv) If the pump casing and suction pipe can be filled with water, this will give a chance of detecting obvious leaks in the system.

(v) Turn the set by hand to make sure that all is free.

(vi) Turn the set under its own power for a few revolutions, to check that the direction of rotation is correct. An arrow cast on the pump casing usually shows the right direction.

(vii) If the installation includes auxiliary machines, run these and check their performance.

354. Starting the Pump. (*a*) *Small Set with Hand Priming.* (i) With the set at rest, *close* the delivery sluice-valve and *open* the priming cock and the vent cock, § 291. In a multi-stage pump the vent-cock on *every* stage must be opened.

(ii) Pour water into the priming funnel until excess water is seen to come out of the air vent. Turn the shaft slowly by hand, to make sure that all air is dislodged from pockets in the impeller. Then shut all cocks.

(iii) Run the pump up to speed without delay. If the pressure-gauge shows that the impeller is duly generating pressure, *gradually* open the main delivery sluice-valve, and leave it set at the opening that gives the desired discharge or pressure. If discharge is controlled by speed-variation, § 242, open the valve fully and speed up the set until it is giving its specified duty.

(*b*) *Large Centrifugal Pump with Evacuating Apparatus.* (i) Shut the delivery sluice-valve, or see that the reflux-valve on the delivery side of the pump is properly closed, § 292.

(ii) Set the exhausting apparatus to work, open the evacuating valve on the pump casing, and watch the gauge-glass, § 294.

(iii) When the gauge-glass shows that the casing is full of water, shut the evacuating valve, stop the evacuating pump, and start the main pump.

(iv) Gradually open the main delivery valve. A by-pass connection on the main valve will facilitate the charging of the delivery pipe, and thereafter, when the pressures on the two faces of the main valve are equalised, the main valve can easily be opened.

The reason for letting the flow build up slowly is this: a *suddenly* opened valve admitting a rush of water into a long, horizontal, empty delivery pipe would compel the pump to yield its maximum discharge against minimum head, possibly throwing a momentary overload on the motive unit. Still more serious consequences may follow if the valve is set some distance away from the pump. Supposing it to be left partially open at the time the pump is started, then the displaced air escaping past it will be succeeded by a rapidly-moving column of water. When this column impinges on the valve face, a most violent and dangerous form of water-hammer will result, § 265.

(c) *Self-priming Pumps.* These call for no special procedure beyond the original filling of the casing with water, § 154.

(d) *Propeller Pumps.* These should never be controlled by a delivery valve, because of the excessive pressure generated and the excessive power absorbed at zero discharge, § 213. The starting routine is correspondingly simplified. If the pump is set below suction level, §§ 327, 328, flow will begin immediately after the set has picked up speed.

(e) *Re-starting the Pump.* When the system has been taken into normal operation, and the pump is to be restarted after a short period of idleness, the starting procedure may be simplified. Supposing, for instance, that the small pump, (*a*) above, feeds into a high-level tank, then the entire delivery system will already be fully charged; the pump can thus be primed directly by slightly opening the delivery valve and opening the vent cock. A small-bore by-pass connection will make matters still easier.

If the pump has been standing for a long time, it is particularly necessary to try to turn it by hand before switching on the power. The rotor may have become rusted or silted up, and quite a strong effort may be needed to break it loose.

355. Routine Attention during Running. Until the new pump has thoroughly settled down to its work, its behaviour should be watched, thus:

(i) Note the pressure-gauge or instrument readings to make sure that liquid is steadily flowing through the system, § 295.

(ii) Feel the glands to guard against overheating. The gland nuts can gently be tightened if everything is running cool.

(iii) If a multi-stage pump has a hydraulic balance-disc, verify that water issues freely from the drain, § 124.

(iv) If there are ring-oiled bearings, open the lids of the oil-wells and make sure that the rings are revolving.

At regular intervals there are routine duties to be carried out:—

(v) Replenish oil or grease in all bearings.

(vi) Change charts on automatic recording instruments, etc., etc.

(vii) Insert new packing-rings in the stuffing-boxes, or put in completely new packing.

Conditions that should not be allowed to occur if at all possible include :—

(viii) Letting the pump run for a long time against closed throttle. The energy continually fed into the liquid will cause a steady rise in temperature, resulting eventually in a dangerous heating of the pump and its contents, § 206.

(ix) Letting the pump run continuously without any liquid in the casing. This may encourage the stuffing-boxes or other components to overheat.

356. Stopping the Pump. It is this operation which, more than any other, may endanger the pump and piping, §§ 263 to 267, 326 to 337. But the normal routine for a manually-controlled centrifugal pumping-set is invariable :—

(i) Gradually close the main delivery valve. If this valve has a by-pass, the by-pass should first be opened, then the main valve closed, and finally the by-pass closed.

(ii) Immediately flow through the pump has ceased, shut off power from the motive unit in the normal way and bring the set to rest.

For other types of pumping-set, refer to the respective paragraphs just specified.

357. Some Possible Faults, (*a*) *General.* Imperfections in pump behaviour, or failure to deliver liquid at all, can be classified thus (it is of course assumed that obvious faults have already been corrected, § 180) :—

(i) *Symptoms.* Although, after the pump has been run up to speed, the pressure-gauge shows that pressure is being generated, yet no liquid flows through the pump.

Fault. Speed too low for intended duty.

Remedy. Increase speed if possible, e.g. by altering size of belt pulleys.

(ii) *Symptoms.* Pump generates stipulated head, but gives insufficient discharge.

Fault. Speed too low, or pump too small, or suction lift excessive.

Remedy. Increase speed, or change pump, or reduce suction lift.

(iii) *Symptoms.* Pump generates excessive head or gives excessive discharge, thereby overloading motive unit.

Fault. Speed too high.

Remedy. Reduce speed : turn down impeller as in § 181.

(iv) *Symptoms.* Pump runs hot or noisily.

Fault. Shafts sprung or out of alignment : faulty entrance conditions, § 76 : glands too tight : excessive suction lift.

Remedy. Correct as indicated.

358. Some Possible Faults, (*b*) *Due to Suction Conditions.* More often than not, unreliable running can be attributed to air leaking into the suction system (*), thus :—

Symptoms. The pump runs normally for a few minutes, and then " loses its water ", i.e. ceases to deliver : unless the attendant keeps a careful eye on his gauges, this may happen without his knowledge. If the pump has a reflux valve on the delivery side, and an open suction pipe, water will gradually drain out of the casing and leave the pump running dry, § 355 (ix).

Cause. (i) The level in the suction well has fallen below the mouth of the suction pipe and allowed air to enter.

(ii) Leaks in the joints of the suction pipe admit air to the pump.

(iii) Air may likewise enter through the main stuffing-boxes if the packing is not properly maintained. If the glands are split, § 86, it is possible for a careless attendant to leave a lower half outside the stuffing-box altogether.

(iv) If the delivery conditions are such that the pump is working under siphonic conditions, § 307, the water-sealing system for the glands may become inoperative and air will seize its chance to get into the pump.

(v) It may happen that the pump draws from an open tank or chamber that is continually replenished by a pipe discharging above the surface level. Large quantities of air bubbles will thereby be entrained, and these can easily be carried across into the pump suction.

(vi) Even if all water enters the suction chamber below surface level, eddies and swirls set up as the water flows round corners or obstructions may likewise entrain air.

(vii) An " air-lock " may have developed in a faultily contrived suction system, § 305.

Remedies. The respective remedies for these various faults are fairly obvious. The leakages must be corrected or, in case (vi), the mouth of the suction pipe must be more deeply

submerged. If the leakage is only slight and, after the loss of water has been observed, it is essential to get the pump to work again without delay, manifestly the entire suction system must be primed anew. An exhausting device may permit this to be done without stopping the pump; but if hand priming is necessary it must be carried out with unusual thoroughness, after the set has been stopped. Especially with multi-stage pumps, it is important to turn the pump slowly by hand and to make sure that water issues from the vent-cock of *each* stage, before starting up the set again.

359. Suction Troubles (*continued*). Air leaks that normally pass unnoticed while the pump is working at full discharge may become apparent if the flow is reduced. Although the normal velocity in the pump passages may be high enough to sweep air bubbles away before they have time to accumulate, yet at lower velocities the concentration of air is sufficient to impair the pressure generated, § 236, and thus to stop the flow altogether.

One type of layout enables leaks to be detected very easily; it is that in which the pump delivers into an open tank or reservoir visible to the attendant. Air bubbles noticeably rising to the surface are a sure sign of potential trouble.

A distinctive symptom of *excessive suction lift*—apart from those mentioned in § 357—is sluggishness of the pump in responding to variations of speed or of head. If the pump is free to obey its own characteristic laws, an increase in speed or a reduction in total head should cause a substantially greater discharge; but this expectation will not be realised if cavitation is in progress, § 245.

360. Faults Developing after Prolonged Service. In course of time we may expect that inevitable wear of the pump parts, and possibly other causes, will make it more and more difficult for the pump to maintain its original level of performance, e.g. :—

(i) The head and efficiency will decline because of increased internal leakage loss and increased frictional resistance, § 258.

(ii) Although the pump may have shown no unmistakable signs of distress, nevertheless cavitation erosion has possibly been ceaselessly in action, with very damaging results, § 257.

(iii) Air leaks may be accentuated by wear on the shaft sleeves, § 81.

(iv) Another cause of excessive gland leakage may be the choking of the water sealing connections, § 85.

(v) If the pump has a foot-valve and strainer, § 290, the effect of allowing the strainer to become choked with rubbish will be equivalent to an excessive suction lift.

(vi) In a multi-stage pump, wear on the balance disc and on the adjacent facing may be responsible for an appreciable axial displacement of the whole rotating element. As the impellers no longer register properly with their respective diffuser rings, excessive energy losses due to eddying and turbulence occur in these regions.

(vii) Undue play in the main bearings may have developed.

361. Repairs and Renewals. Here are some considerations that may come up for study when trying to decide when the worn pump must be repaired or replaced :—

(i) The conditions of constant dead head and constant speed, § 240, allow least latitude. Especially if the head-discharge characteristic is flat, quite a small amount of wear will pull the characteristic down sufficiently, Fig. 185, to reduce the discharge quite seriously. Before very much longer, the pump might cease to deliver altogether. There is no question of deferring repairs here.

(ii) If the pump speed is variable, and if the motive unit has a sufficient reserve of power, the pump can be forced to keep up to its work by progressively raising the speed. But this does nothing to check the excessive power input the pump demands.

(iii) While in course of time the pump is getting worse, we have to remember that the motive unit is not getting any better. An oil engine, for instance, might lose its capacity even more rapidly than the pump ; and there may be a definite limit to these concurrent tendencies.

(iv) In high-grade installations fully equipped with instruments, § 295, it is possible to work out accurately how much the decline in pump efficiency is costing, and to balance this against the cost of renewals, § 298.

(v) If the conditions of service allow the pump to stand idle at stated intervals, naturally the maintenance staff will take

advantage of these periods to examine the installation, even though the pump shows no manifest sign of deterioration.

(vi) A group of constant-speed pumps working in parallel should not be reconditioned piecemeal. If half-measures are tried—if one pump is repaired and the others left as they are—the result may be that the favoured pump will have to do a disproportionate share of the work, while the worn pumps do even less than they did before, § 243 (ii). In extreme cases, some of the liquid pumped by the repaired unit may run *backwards* through the remaining units.

362. Reconditioning the Pump. Possibilities of improvement have been foreshadowed in §§ 259, 360. If the pump has been suitably designed, parts that can advantageously be replaced are :—

Wearing rings on rotor or in casing.
Shaft sleeves.
Glands and neck-bushes.
Balance-disc or its facing.

Eroded metal in the casing or on the rotor may be built up again by welding. If the rotor shows signs of material damage, it is worth while studying the question of fitting a new rotor, which may be better suited to existing service conditions than the original one. Possibly since the set was first installed, the ruling head or discharge demanded of the pump may have altered, and a redesigned rotor might thus be a very economical purchase.

PART E

ILLUSTRATIVE EXAMPLES

EXAMPLE 1.

To calculate the ideal energy gain, blade angles, etc., for a given centrifugal pump, § 13.

Data :—

Dimensions and velocities are those used in Figs. 4, 5 (iii), 7, 8, and 9 (*b*), viz. :—

d_1 = inner diameter of impeller	=	1 ft.
d_2 = outer diameter of impeller	=	2 ft.
b_1 = width at inner rim	=	0·2 ft.
b_2 = width at outer rim	=	0·1 ft.
v_1 = inner rim speed	=	15 ft./sec.
v_2 = outer rim speed	=	30 ft./sec.
Y = radial flow component (constant)	=	8 ft./sec.
V_2 = outlet tangential velocity component or velocity of whirl	=	20 ft./sec.

Method :—

From equation (1-2),

$$E = \text{energy input} = \frac{V_2 v_2}{g} = \frac{20 \times 30}{32 \cdot 2} = 18 \cdot 6 \text{ ft. lb./lb.}$$

$$\text{Energy wasted} = \frac{U_2{}^2}{2g} = \frac{Y_2{}^2 + V_2{}^2}{2g} = \frac{8^2 + 20^2}{64 \cdot 4} = 7 \cdot 2 \text{ ft. lb./lb.}$$

Therefore *useful energy* given to water $= 18 \cdot 6 - 7 \cdot 2 =$ ***11·4 ft.***

This value 11·4 represents either the energy in foot-pounds received by each pound of water, or the ideal height H through which the water is lifted, Fig. 8.

$$\textit{Outlet blade angle } \gamma = \tan^{-1} \frac{Y_2}{v_2 - V_2} = \tan^{-1} \frac{8}{30 - 20}$$
$$= 38 \cdot 6^\circ.$$

$$\textit{Inlet blade angle } \beta = \tan^{-1} Y_1/v_1 = \tan^{-1} 8/15$$
$$= 28^\circ.$$

If the blades are shaped accordingly, the water will slide smoothly on to them at inlet, and will receive from them the stipulated energy increment.

Discharge $q = \pi d_2 b_2 Y_2 = 3 \cdot 14 \times 2 \times 0 \cdot 1 \times 8 = 5$ cu. ft./sec.

Weight per second $= W = qw = 5 \times 62{\cdot}4 = 313$ lb./sec.

Useful power output $= \dfrac{WH}{k_p} = \dfrac{313 \times 11{\cdot}4}{550} = \mathit{6{\cdot}49\ h.p.}$

Power input $= \dfrac{WE}{k_p} = \dfrac{313 \times 18{\cdot}6}{550} = \mathit{10{\cdot}56\ h.p.}$

Ideal efficiency of impeller $= 6{\cdot}49/10{\cdot}56 = \mathit{0{\cdot}61}$.

Rotational speed $= N = \dfrac{60 \times v_2}{\pi d_2} = \dfrac{60 \times 30}{3{\cdot}14 \times 2} = \mathit{287\ r.p.m.}$

EXAMPLE 2.

To estimate the pressure distribution in a revolving impeller, and thereby to deduce the energy given to the water, § 20.

Data :—

The impeller has the same size and speed as in Example 1 above ; but it now has twelve radial blades instead of curved blades, Fig. 13.

Method :—

Angular velocity $= v_2/r_2 = 30/1 = 30$ rad./sec.

$$\left(\text{or } \omega = \frac{2\pi N}{60} = \frac{6{\cdot}28 \times 287}{60} = 30 \text{ rad./sec.}\right)$$

Tangential acceleration $\alpha = 2Y\omega$ (from equation 2-1)

$= 2 \times 8 \times 30 = 480$ ft./sec./sec.

(*Note* that even in this slow-speed, low-head machine, the tangential acceleration is already fifteen times as great as the acceleration of gravity.)

Difference of pressure-head due to forced vortex flow :—

This can be found from the expression :—

$$h_c = \frac{v_2^2 - v_1^2}{2g} = \frac{30^2 - 15^2}{64{\cdot}4} = \mathit{10{\cdot}5\ ft.\ head.}$$

(see Fig. 13.)

Differential pressure-head across blade :—

From equation (2-2), when $r = 1$ (at outer rim)

$$h_d = \frac{1}{32{\cdot}2} \times \frac{8 \times (3{\cdot}14)^2}{60} \cdot \frac{1 \times 8 \times 287}{12}$$
$$= \mathit{7{\cdot}8\ ft.\ head.}$$

Similarly when $r = 0{\cdot}5$ (at inner rim), $h_d = 3{\cdot}9$ *ft. head.*

Dynamic thrust on blade $= P_d = \int w \,.\, h_d \,.\, b \,.\, dr$.

But in this instance, $h_d = 7{\cdot}8\ r$ (from above), and $b = 0{\cdot}1/r$ (since Y is constant).

Therefore $$P_d = \int_{r=0{\cdot}5}^{r=1} 62{\cdot}4 \times 7{\cdot}8\, r \times 0{\cdot}1/r \times dr,$$
$$= \mathit{24{\cdot}3\ lb.\ per\ blade.}$$

Torque on blade $= T \quad = \int_{r=0\cdot5}^{r=1} w \,.\, h_d \,.\, b \,.\, dr \,.\, r,$

$= 18\cdot2$ *ft. lb. per blade.*

Work done per second in imparting torque to 12 blades

$= T\omega n = 18\cdot2 \times 30 \times 12 =$ *6550* ft. lb./sec.

Energy imparted to water per second

$= \frac{W}{g}\left(V_2v_2 - V_1v_1\right)$ (from equation (1-2), modified to take into account whirl velocity at entry),

$= 313\cdot\frac{(30 \times 30 - 15 \times 15)}{32\cdot2} =$ *6550 ft. lb./sec.*

The two values agree, as they ought to do in these ideal circumstances.

EXAMPLE 3.

To estimate the average loading on an ideal impeller blade, § 26.

Data :—

As in Example 1 and Fig. 8.

Method :—

a_t = projected area of blade = $0\cdot14 \times 0\cdot5 = 0\cdot07$ sq. ft.
r_g = radius of C.G. of projected area = $0\cdot73$ ft.
P_t = tangential component of dynamic thrust per blade,

$= \frac{\text{Input power} \times k_p}{n \,.\, r_g \,.\, \omega} = \frac{10\cdot56 \times 550}{8 \times 0\cdot73 \times 30} = 33$ lb.

Average differential pressure $= P_t/a_t = 33/0\cdot07$

$=$ *475 lb./sq. ft.* $=$ *3·3 lb./sq. in.*

Average differential pressure-head $= h_d \quad 475/62\cdot4$

$-$ *7·6 ft.*

Relative blade loading ϵ $= h_d/H$

$= 7\cdot6/11\cdot4$

$=$ *0·73.*

(The difference in the shape and number of blades accounts for the difference in dynamic thrust recorded in Examples 2 and 3.)

EXAMPLE 4.

To compute energy loss, and recovery of pressure-head, in simple straight-line recuperators, § 43.

Data :—

Referring to Fig. 25, let

d_1 = inlet diameter = 6 in.
d_2 = outlet diameter = 12 in.
v_1 = inlet velocity = 20 ft./sec.
v_2 = outlet velocity = 5 ft./sec.

Method :—

(*a*) *Sudden enlargement.*

The usual expression for *energy loss* is :—

$$h_l = \frac{(v_1 - v_2)^2}{2g} = \frac{(15)^2}{64 \cdot 4} = 3 \cdot 50 \text{ ft. lb./lb.}$$

Recuperation of pressure-head $= h_g$

$$= \frac{v_1^2}{2g} - h_l - \frac{v_2^2}{2g} = 6 \cdot 22 - 3 \cdot 50 - 0 \cdot 39 = 2 \cdot 33 \text{ ft.}$$

$$\textit{Efficiency} \text{ of conversion} = \frac{\text{regain of head}}{\text{velocity energy input}} = \frac{h_g}{v_1^2/2g}$$

$$= \frac{2 \cdot 33}{6 \cdot 22} = \mathit{0 \cdot 37.}$$

(*b*) *Gradual or conical enlargement.*

Minimum energy *loss* occurs with the gradual taper shown in Fig. 25 (*b*), and it has the approximate value :—

$$h_l = 0 \cdot 14 \frac{(v_1 - v_2)^2}{2g} = 0 \cdot 49 \text{ ft. lb./lb.}$$

$$\textit{Recuperation} \text{ of pressure-head} = 6 \cdot 22 - 0 \cdot 49 - 0 \cdot 39$$
$$= 5 \cdot 34 \text{ ft.}$$

$$\textit{Efficiency} \text{ of conversion} = \frac{5 \cdot 34}{6 \cdot 22} = \mathit{0 \cdot 86.}$$

EXAMPLE 5.

To estimate the ideal regain of pressure-head in a vortex chamber or whirlpool chamber, § 43.

Data :—

Inner diameter 2 ft.
Outer diameter = 4 ft.
Axial width (uniform) = 0·1 ft. } as in Fig. 26.

Water enters the inner periphery of the chamber with the velocity components with which it leaves the impeller in Fig. 8, viz., velocity of flow, $Y_2 = 8$ ft./sec., velocity of whirl, $V_2 = 20$ ft./sec.

Method :—

Radial velocity component at chamber *outlet* = 4 ft./sec.

Therefore ideal recuperation of pressure-head due to fall in *radial* velocity, § 8, $= h_{gr} = \dfrac{8^2}{64 \cdot 4} - \dfrac{4^2}{64 \cdot 4} = \mathit{0 \cdot 75 \; ft.}$

Tangential component at outlet = 10 ft./sec., because of the free vortex relationship Vr = constant.

Ideal recuperation of pressure-head due to fall in tangential velocity

$$= h_{gt} = \frac{20^2}{64 \cdot 4} - \frac{10^2}{64 \cdot 4} = 4 \cdot 66 \text{ ft.}$$

Ideal gross gain in pressure head $= h_{gr} + h_{gt} = 0{\cdot}75 + 4{\cdot}66$
$= \mathit{5{\cdot}41\ ft.}$

EXAMPLE 6.

To plot ideal shape of volute casing, on the assumption that free vortex flow prevails in it, §§ 46, 88.

Data :—

As in Fig. 8, viz., whirl velocity component of water leaving impeller = 20 ft./sec. Discharge = 5 cu. ft./sec.

Method :—

First compute the volute diameter at the sections (1), (2) and (3), Fig. 28, on the original basis of uniform velocity distribution, § 45, thus :—

Section.	Discharge Past Section.	Mean Velocity.	Cross-sectional Area.	Diameter.
	cub. ft./sec.	ft./sec.	sq. ft.	ft.
(1)	1·25	20	0·0625	0·28
(2)	2·5	20	0·125	0·40
(3)	3·75	20	0·187	0·49
(4)	5·0	20	0·25	0·56

Next apply the correction for free vortex flow thus :—

If d = diameter of a curved circular passage,
r_3 = inner radius of curvature,
v_w = *mean* tangential velocity in passage = discharge/area,
V_3 = actual tangential velocity at inner radius (see diagram (1) overleaf),
m = ratio V_3/v_w.

Then it can be shown that the ratio $m = \frac{V_3}{v_w}$ has the value :—

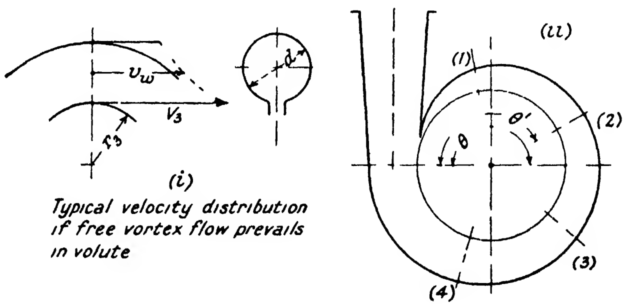
$$= 1 \cdot 02 + 0 \cdot 43 \frac{d}{r_3} \text{ (approx.).}$$

Inserting in this expression the figures just obtained, we find that at the respective cross-sections the values of m are respectively : (1) 1·13 ; (2) 1·19 ; (3) 1·23 ; (4) 1·26. This means that at section (2), for instance, water leaving the impeller with a whirl component of 20 ft./sec would find itself in a stream of liquid moving round the casing with a velocity of 20 × 1·19 = 23·8 ft./sec. But to ensure minimum energy loss these two velocities should be equal, viz., $V_2 = V_3$, in just the same way that a passenger alighting from a

moving vehicle knows that for a few moments he should run alongside the vehicle *without change of speed.*

It is true that the correction could be made by enlarging the area of the volute at the selected section, but a quicker method is as follows : Let us keep the same area, but move the section to a point on the volute at which the discharge and therefore the velocities have been reduced in the desired ratio.

If θ is the angular position of the section in relation to the zero point or origin of the volute, diagram (ii), and θ' is the corrected or new position of the section, then clearly $\theta' = \dfrac{\theta}{m}$; because the discharge flowing past any section is directly proportional to θ. Instead, therefore, of spacing the sections at angles respectively of 90°, 180°, 270° and 360°, as in Fig 28, we must swing the sections backwards until the angles are respectively 80°, 152°, 220° and 286°, as in diagram (ii).



Typical velocity distribution if free vortex flow prevails in volute

Although the resulting increase in diameter *at a given angular position* may appear to be inconsiderable, yet the corresponding advantage in respect of reduced frictional loss is well worth having : for this loss varies inversely as the *fifth* power of the diameter. It is for this reason that the casing may profitably be enlarged even more than the foregoing method of correction indicates, § 88.

EXAMPLE 7.

To compute the specific speed of a pump, and to describe its shape and performance in non-dimensional terms, § 59.

Data :—

The same figures as in Example 1 will be used, except that now the effect of the recuperator can be taken into account. The total head generated by the pump may therefore be taken as $H = 15{\cdot}6$ ft.

Method :—

Width ratio $\quad \lambda = b_2/d_2 = (0{\cdot}1)/2 = \mathit{0{\cdot}05}.$

Speed ratio $\quad \phi = \dfrac{v_2}{\sqrt{2gH}} = \dfrac{30}{8{\cdot}03\sqrt{15{\cdot}6}} = \mathit{0{\cdot}95}.$

Flow ratio $\psi = \frac{Y_2}{\sqrt{2gH}} = \frac{8}{31\cdot7} = \mathit{0\cdot25}.$

Shape number $n_s = 950\phi\sqrt{\lambda\psi}.$

$$= 950 \times 0\cdot95 \sqrt{0\cdot05 \times 0\cdot25} = \mathit{101}.$$

Alternatively $n_s = \frac{1000n\sqrt{Q}}{(gH)^{\frac{3}{4}}} = \frac{1000 \times \frac{287}{60}\sqrt{5}}{(32\cdot2 \times 15\cdot6)^{\frac{3}{4}}}.$

The value of $(32\cdot2 \times 15\cdot6)^{\frac{3}{4}} = (502)^{\frac{3}{4}}$ can be found on the slide rule thus :—

$$\sqrt{502} \times \sqrt{\sqrt{502}} = 22\cdot4 \times \sqrt{22\cdot4} = 106.$$

Inserting in the general expression, we find $n_s = \mathit{101}$.

To compute the specific speed, we find Q in gallons per minute $= 5 \times 374 = 1870$ gall./min.

Nominal specific speed $N_{sn} = \frac{N\sqrt{Q}}{H^{\frac{3}{4}}} = \frac{287\sqrt{1870}}{\sqrt{15\cdot6} \times \sqrt[4]{15\cdot6}}$

$$= 1580$$

True specific speed $N_s = 0\cdot0174\, N_{sn} = 27\cdot5$ (foot).

These values can rapidly be read off from the chart facing p. 480.

Characteristic head number $h_c = \frac{gH}{n^2 d_2^{\,2}}$

$$= \frac{32\cdot2 \times 15\cdot6}{4\cdot78^2 \times 2^2} = \mathit{5\cdot50}.$$

Characteristic discharge number $q_c = \frac{q}{n\, d_2^{\,3}} = \frac{5}{4\cdot78 \times 8} = \mathit{0\cdot13}.$

EXAMPLE 8.

To choose a suitable type of pump for a specified duty, § 64.

Data :—

(*a*) Liquid : cold water.
Discharge : 300 gall /min.
Head : 65 ft.

Method :—

A centrifugal pump is clearly indicated. From Fig. 52, a shape number of about 100 might be suitable. When cold water is concerned, the formula for shape number can be put in the form :—

$$n_s = 0\cdot064. \frac{\text{Speed in r.p.m.}\sqrt{\text{gallons per minute}}}{(\text{head in feet})^{\frac{3}{4}}}$$

from which N = speed of pump = *2050 r.p.m.*

(*b*) If the pump were to be direct-coupled to a motor running at 1450 r.p.m., could a centrifugal pump still be used ?

Inserting appropriate values in the expression for shape number, we find :—

$$n_s = 0{\cdot}064 \, . \, \frac{1450\sqrt{300}}{(65)^{\frac{3}{4}}} = \mathit{70}$$

which is quite a suitable figure.

Again the calculations can be much simplified by the chart facing p. 480.

EXAMPLE 9.

To select a suitable type and speed for a low-lift pump, § 64.

Data :—

Discharge $= 3$ cu. m./sec.
Head $= 2{\cdot}4$ m.

Method :—

An axial-flow pump could provisionally be chosen, having a shape number of about 600. The speed can be found from the standard shape number formula, thus :—

$$600 = \frac{\frac{1000\,N}{60}\sqrt{2{\cdot}4}}{(9{\cdot}8 \times 3)^{\frac{3}{4}}}, \text{ from which } N = \text{about } 300 \ \textit{r.p.m.}$$

If a screw pump of shape number 350 were preferred, it would have to run at *175 r.p.m.*

EXAMPLE 10.

To estimate the unbalanced thrust on a side-inlet impeller, § 74.

Data :—

Head	= 130 ft.
Outer impeller diameter	= 15 in.
Inner diameter	= 7 in.
Shaft diameter	= $1\frac{3}{4}$ in.
Rim speed	= 90 ft./sec.
Axial velocity of flow	= 15 ft./sec.

Method :—

The values to be inserted in equation (7-1) are :—

$w = 62{\cdot}4$ lb./cu. ft. ; $H = 130$ ft. ; $r_2 = 0{\cdot}625$ ft. ; $r_1 = 0{\cdot}292$ ft. ; $r_0 = 0{\cdot}073$ ft., giving the result :—

$$P_a = \text{unbalanced static thrust} = \textit{1605 lb.}$$

In estimating the dynamic thrust by the formula of § 75 (*b*), we use the values

$$Y = 15 \text{ ft./sec.,}$$

and $$W = Qw = 3{\cdot}14\,(0{\cdot}292^2 - 0{\cdot}073^2) \times 15 \times 62{\cdot}4$$
$$= 236 \text{ lb./sec.}$$

Hence $$P_a' = \frac{236}{32{\cdot}2} \times 15 = \textit{110 lb.}$$

Since the dynamic thrust acts in the opposite direction to the static thrust, the net unbalanced axial thrust = 1605 − 110 = about *1500 lb.*

This is quite an appreciable amount, especially as the shaft speed will be about 1400 r.p.m. It will be noted that the dynamic thrust is trivial in comparison with the static thrust.

EXAMPLE 11.

To estimate the leakage loss in a side-inlet centrifugal pump, § 83.

Data :—

Impeller as in Example 10.

Axial length of leakage passage = $\frac{5}{8}$ in.

Radial width of leakage passage (assumed uniform and parallel) = 0·02 in.

Value of coefficient "f" = 0·01.

Method :—

According to § 83, difference of pressure-head creating leakage flow is represented by

$$H_l = 130 - \frac{\left(\frac{90}{2}\right)^2}{64{\cdot}4}\left[1 - \left(\frac{0{\cdot}292}{0{\cdot}625}\right)^2\right]$$

$$= 105 \text{ ft.}$$

Also H_l = energy loss as water flows through leakage space

$$= \left[\frac{0{\cdot}5 + \frac{2 \times 0{\cdot}01 \times 0{\cdot}625}{0{\cdot}02} + 1}{64{\cdot}4}\right] \times v_l^2$$

$$= (0{\cdot}5 + 0{\cdot}625 + 1)\,\frac{v_l^2}{64{\cdot}4}.$$

Equating the two values of H_l, we find

v_l = velocity of flow through leakage space = *56·5 ft./sec.*

(It is significant that of the three items that constitute the hydraulic resistance of the passage, § 82, the frictional loss h_f is not the preponderant one. If it were neglected altogether, the estimated value of the leakage velocity would be increased by less than 20 per cent.)

Leakage area = $6{\cdot}28 \times 0{\cdot}292 \times (0{\cdot}02)/12 = 0{\cdot}00305$ sq. ft.

Therefore leakage flow $q_l = 56{\cdot}5 \times 0{\cdot}00305$ = *0·17 cu. ft./sec.*

Total flow through impeller = 236/62·4 = 3·78 cu. ft./sec.

Therefore percentage leakage loss = $\frac{0{\cdot}17}{3{\cdot}78} \times 100$ = *4·5 per cent.*

To find the value of the coefficient of discharge C_d in the expression in § 83, we have :—

$$0{\cdot}17 = C_d \times (0{\cdot}00305) \times 8{\cdot}03 \times 10{\cdot}2,$$

from which $C_d = 0{\cdot}68$.

EXAMPLE 12.

To establish the main dimensions of a split-casing centrifugal pump, §§ 90 to 94.

Data :—

Effective head = 60 ft.
Discharge = 400 gall./min.
Speed = 1450 r.p.m.

Method :—

(i) First check the specific speed. From the chart facing p. 480, we find that this is 23 (foot), which is quite permissible. The corresponding shape number $= 23/0{\cdot}273 = 84$. According to Fig. 52, this value is well within safe limits.

(ii) *Discharge* in cubic feet per second $= 400/374 = 1{\cdot}07$.

(iii) *Output power* $= P_w = \dfrac{1{\cdot}07 \times 62{\cdot}3 \times 60}{550} = 7{\cdot}3$.

(iv) *Gross efficiency* = about 73 per cent. (from Fig. 53).

(v) *Power input* $= P_s = P_w/0{\cdot}73 = 10$.

(vi) *Impeller dimensions.* From Fig. 54, read off values for double-inlet impellers corresponding to speed number $n_s = 84$, viz. :—

Speed ratio = 1·00.
Diameter ratio = 0·45.
Width ratio = 0·085.

Now v_2 = outer rim speed $= \phi\sqrt{2gH_e}$
$= \dfrac{\pi d_2 N}{60}$,

from which $1{\cdot}00 \times 8{\cdot}03\sqrt{60} = \dfrac{3{\cdot}14 \times d_2 \times 1450}{60}$.

Thus d_2 = outer diameter = *0·82 ft.*
b_2 = outer width $= \lambda d_2 = 0{\cdot}085 \times 0{\cdot}82 =$ *0·07ft.*
d_1 = inner diameter $= d_2 \times$ diameter ratio
$= 0{\cdot}82 \times 0{\cdot}45 =$ *0·37 ft.*
b = inner width $= 0{\cdot}07/0{\cdot}45 =$ *0·155 ft.*

(vii) *Impeller blades. Number n* may be taken as 7. *Thickness t* may be taken as 4 mm.

From the expression $\dfrac{1-\eta_h}{1-\eta_m} = 0{\cdot}5$, find the value of the *hydraulic efficiency* $= \eta_h = 0{\cdot}87$.

From the expression $H_e = \eta_h \cdot \dfrac{V_n v_2}{g}$, or

$$60 = 0{\cdot}87 \cdot \frac{V_n \times 62{\cdot}1}{32{\cdot}2},$$

find the value of the *true outlet whirl component*
V_n = *35·7 ft./sec.*

From the expression $V_\infty = V_n\left(1 + \frac{3}{7}\right)$, find

$$V_\infty = \textit{ideal whirl component} = 51{\cdot}0 \text{ ft./sec.}$$

From Fig. 54, estimate the value of the *flow ratio*

$$\psi = Y_2/\sqrt{2gH_e} = 0{\cdot}105,$$

from which :—

$$Y_2 = \textit{radial flow component} = 6{\cdot}52 \text{ ft./sec.}$$

From the expression $\cot\gamma = \dfrac{v_2 - V_\infty}{Y_2} = \dfrac{62{\cdot}1 - 51{\cdot}0}{6{\cdot}52}$, find $\gamma = \textit{outlet blade angle} = 30^\circ\ 23'$.

From the expression $\tan\beta = Y_1/v_1$, or $\dfrac{6{\cdot}52}{27{\cdot}9}$, find

$$\beta = \textit{inlet blade angle} = 13^\circ\ 8'.$$

(viii) *Correction to impeller width.* In order to allow a margin for wear, the *leakage loss* q_l may be taken as 10 per cent. of the net flow Q. By the method of § 94, it is found that :—

Corrected inlet width $= 0{\cdot}26$ ft.
Corrected outlet width $= 0{\cdot}083$ ft.

As the inlet angle β appears to be rather small, it might be desirable to increase it slightly and reduce the inlet width to correspond.

(ix) *Casing dimensions.* In the volute, the mean velocity v_w may be taken as $\frac{2}{3}$ of the true whirl component of the water leaving the impeller, viz.

$$v_w = 0{\cdot}67 \times 35{\cdot}7, \\ = 23{\cdot}8 \text{ ft./sec.}$$

By the method of Example 6, the respective volute diameters at sections spaced respectively 90, 180, 270, and 360 from the origin are found to be 0·12 ft., 0·17 ft., 0·21 ft., and 0·24 ft. There is of course no need to apply the correction for free vortex flow, because this has already been done in a crude fashion by taking v_w as $\frac{2}{3}$ of V_n.

In the *inlet and outlet branches,* the mean velocity may provisionally be equated to the velocity of flow in the impeller, viz., 6·52 ft./sec. This corresponds to a diameter of 0·46 ft. or $5\frac{1}{2}$ in. To conform with the usual practice of making the suction branch bigger than the delivery branch, we may choose the nearest standard size thus :—

Either : Suction 5 in., Delivery 4 in.
or : Suction 6 in., Delivery 5 in.

The smaller size gives a better rate of taper in the conical outlet branch connecting the volute outlet to the actual delivery branch.

The general proportions of this pump are shown in Fig. 58.

EXAMPLE 13.

To estimate the main dimensions of a screw-pump, § 106.

Data :—

Effective head = 22 ft.
Discharge = 6500 gall./min.
Speed : as required.

Method :—

(i) *Speed*. As there are no restrictions on the value of the shape number, it can arbitrarily be given any reasonable value—say 350. Inserting known values in the proper formula, as in Example 8, or by the use of the graphs, we find

$$N = \text{rotational speed} = 692 \text{ r.p.m.}$$

The nearest round number would be *700 r p.m.*, but if it were desired to couple the pump directly to an A.C. electric motor, then a speed of 730 would be equally satisfactory. For present purposes the value $N = 700$ r.p.m. may be accepted ; it corresponds to a shape number of $n_s = 354$

(ii) *Power, etc.* $P_w = \text{power output} = \dfrac{6500 \times 10 \times 22}{33{,}000}$

$$= 43{\cdot}3 \text{ h p.}$$

The value of the gross efficiency may be taken as $\eta_m = 0{\cdot}82$, from which $P_s = \text{power input} = 43{\cdot}3/0{\cdot}82 = 53{\cdot}0$ h.p.

(iii) *Rotor proportions*. The *speed ratio* at inlet can be taken as $\phi = 1{\cdot}25$; and since $n_s = 475\phi\sqrt{\psi}$, we find ψ (nominal) $= 0{\cdot}355$.

Now $Q = 6500/374 = 17{\cdot}4$ cu. ft./sec., and

$$\psi = \frac{Q\Big/\left(\frac{\pi}{4}d_1^{\,2}\right)}{\sqrt{2gH_e}},$$

from which d_1 = inlet diameter = 1·285 ft.
Taking $d_{2m}/d_1 = 1{\cdot}05$, then $d_{2m} = 1{\cdot}350$ ft.
Taking $\lambda = 0{\cdot}27$, then $b_{2m} = 0{\cdot}27 \times 1{\cdot}350 = 0{\cdot}365$ ft.
Taking $d_{2a}/d_{2b} = 0{\cdot}8$, then $d_{2b} = 1{\cdot}50$ ft., and $d_{2a} = 1{\cdot}20$ ft.

On sketching the rotor outline, and scaling off remaining values, it appears that :—

$$b_1 = 0{\cdot}45 \text{ ft.}, \ d_{1m} = 0{\cdot}82 \text{ ft.}, \ b_2 = 0{\cdot}30 \text{ ft.}$$

The mean meridional velocity at inlet $= Y_{m2}$

$$= \frac{17{\cdot}4}{3{\cdot}14 \times 0{\cdot}82 \times 0{\cdot}45} = 15{\cdot}0 \text{ ft./sec.}$$

Similarly, mean meridional velocity at outlet $= Y_{m2} = 13{\cdot}6$ ft./sec. With this information, the blades may now be set out.

(iv) *Casing*. The diameter of the *inlet branch* is fixed by that of the rotor, viz., 1·285 ft. = $15\frac{1}{2}$ in. The nearest whole

number is *16 in.* In these pumps the delivery branch is often made *greater* than the suction branch. In this instance a convenient figure is *18 in.* diameter. The corresponding outlet velocity is 10 ft./sec., which tallies well enough with the mean outlet whirl velocity component $V_n = 16{\cdot}3$ ft./sec.; the opportunities for recuperation would be quite favourable.

If Fig. 64 were regarded as drawn to a scale of 1/20, it would give a fair representation of this pump.

EXAMPLE 14.

To estimate the main dimensions of a propeller pump, § 109.

Data :—

Effective head $= 3{\cdot}8$ m.
Discharge $= 1250$ lit./sec.
Speed $= 500$ r.p.m.

Method :—

(i) Check the value of the shape number. It works out as 619, which is permissible.

(ii) Assuming a gross efficiency of $\eta_m = 0{\cdot}83$, we find that

$$P_w = \text{output power} = 63{\cdot}3 \text{ h.p., and}$$
$$P_s = \text{input power} = 76{\cdot}3 \text{ h.p.}$$

(iii) Assuming the values ϕ = speed ratio $= 2{\cdot}2$,

$$\text{and } d_a/d_b = 0{\cdot}55,$$

we use the expression $619 = 475 \times 2{\cdot}2\sqrt{\psi}\sqrt{1 - (0{\cdot}55)^2}$ to find ψ = flow ratio

$= 0{\cdot}50$, which is permissible.

(iv) Since $v_{2b} = \phi\sqrt{2gH_e} = \dfrac{3{\cdot}14 \times d_b \times 500}{60}$, we find

$$d_b = \text{outer diameter of rotor} = 0{\cdot}726 \text{ m., and}$$
$$d_a = 0{\cdot}55\, d_b = 0{\cdot}40 \text{ m.} = \text{inner diameter.}$$

Thus v_{2b} = outer rim speed $= 19{\cdot}0$ m./sec.
v_{2a} = inner rim speed $= 10{\cdot}4$ m./sec.
Y_a = velocity of flow $= 4{\cdot}33$ m./sec.

(v) Because of the uncertainty in assessing the value of the ratio V_n/V_∞, it may suffice here to give ideal values of the blade angles, thus :—

If η_h = hydraulic efficiency $= 0{\cdot}88$, then
V_{nb} = whirl component at outer rim $= 2{\cdot}22$ m./sec., and
V_{na} = whirl component at inner rim $= 4{\cdot}05$ m./sec.
Also $\tan \beta_b = Y_a/v_b = 4{\cdot}33/19{\cdot}0 = 0{\cdot}228$, from which
β = inlet blade angle at outer radius $= 12°\ 50'$.
Similarly $\tan \gamma_b = Y_a/(v_b - V_{2b}) = 4{\cdot}33/(19{\cdot}00 - 2{\cdot}22)$
$= 0{\cdot}258$

from which γ_b = outlet blade angle at outer radius
= 14°28′.

In turn, β_a = inlet blade angle at inner radius = 22°32′,
γ_a = outlet blade angle at inner radius = 34°8′.

The true values of γ_a and γ_b would probably be rather greater than the ideal ones just given. If it is preferred to estimate the angles graphically, the typical shape of velocity diagrams is shown in Fig. 23 (i).

(vi) *Axial hydraulic thrust.* The crude methods of §§ 26 and 37 provide quite a useful estimate of the value of the gross axial thrust on the rotor blades, thus :—

Gross power input to pump = P_s = 76·3 h.p.
Corresponding energy per sec. = 76·3 × 75 = 5720 kg. m./sec.
Mean blade radius = r_g, (Fig. 23 (ii)) = 0·282 m.
Mean blade tangential velocity = 14·7 m./sec.
If P_t = gross tangential thrust on *all* blades, then energy per sec. = 5720 = P_t × 14·7, from which P_t = 390 kg.
Mean blade angle = mean of $\beta_a\, \beta_b\, \gamma_a\, \gamma_b$ = 21° = $\tan^{-1}(P_t/P_a)$.

$$\text{Therefore } P_a = \text{total axial thrust} = \frac{390}{\tan 21^\circ} = 1015 \text{ kg.}$$

(say) = rather more than 1 ton.

If the pump were set vertically, the gross load to be carried by the thrust bearing would naturally be the sum of P_a and the weight of the rotating element (rotor, shaft, etc.).

(vii) *Recuperator.*
Finally, $\tan \delta_b$ = 4·33/2·22 = 1·95, from which guide blade angle at outer radius = 63°
and $\tan \delta_a$ = 4·33/4·05 = 1·07, from which guide blade angle at inner radius = 47°.

EXAMPLE 15.

To estimate the main dimensions of a multi-stage centrifugal pump, § 125.

Data :—

Purpose : Boiler-feed pump.
Pressure difference to be generated = 520 lb./sq. in.
Discharge = 280,000 lb./hour.
Speed = 2900 r.p.m.
Water temperature = 250° F.

Method :—

(i) The data must first be put into standardised form, thus :—
w = density of water at 250° F. = 58·8 lb./cu. ft.

Effective head $H_e = \dfrac{520 \times 144}{58{\cdot}8} = 1274$ ft.

Discharge $Q = \dfrac{280000}{3600 \times 58{\cdot}8} = 1{\cdot}322$ cu. ft./sec.

(ii) Choosing a provisional value for the impeller shape number of 60, and inserting this value in the formula of § 125, we find that

$$m = \text{number of stages} = 4{\cdot}54.$$

The actual number will therefore be either 4 or 5. A 4-stage pump would be cheaper, but a 5-stage pump would be more efficient. The latter may be accepted, viz., $m = 5$.
It follows that H_{es} = head per stage = 1274/5 = 255 ft.

$$n_s = \text{actual shape number} = 65.$$

(iii) Power output per stage $= \dfrac{1{\cdot}322 \times 58{\cdot}8 \times 255}{550} = 36{\cdot}1.$

From Fig. 53, corresponding gross efficiency = 0·78. Efficiency of multi-stage pump η_m can thus be assessed at 0·78 − 0·04 = 0·74.

Power input to pump $P_s = \dfrac{5 \times 36{\cdot}1}{0{\cdot}74} = 244$ h.p.

(iv) From Fig. 54, value of speed ratio ϕ is 0·95. Since

$$v_2 = 0{\cdot}95\sqrt{64{\cdot}4 \times 255} = \frac{3{\cdot}14 \times d_2 \times 2900}{60},$$

we find d_2 = impeller diameter = 0·80 ft. Other impeller details are established as in §§ 90 to 95. As V_n = true outlet whirl component = about 77 ft./sec., and δ = diffuser blade angle = $\tan^{-1}(Y_2/V_n)$, we find $\delta = 8\tfrac{1}{2}°$ ideally, or about 10° actually.

(v) Outside diameter of diffuser discs = $d_3 = 1{\cdot}6 \times 0{\cdot}8 = 1{\cdot}28$ ft. (about).
Axial pitch of impellers $l_p = 0{\cdot}4 \times 0{\cdot}8 = 0{\cdot}32$ ft. (about).
If Fig. 82 were regarded as a drawing to a scale of $\frac{1}{14}$, it would give a rough impression of the size and proportions of this pump.

EXAMPLE 16.

To assess the inter-stage leakage in a multi-stage pump, § 120.

Data :—

Pump as Example 15.
Impeller boss diameter = 0·22 ft.
Radial clearance between boss and neck-bush = 0·015 in.

Method :—

By studying a sectional diagram such as Fig. 84, it will be seen that the pressure-difference creating leakage flow is made up of two

items : (i) the centrifugal or forced-vortex pressure-head in the space between the impeller shroud and the fixed diaphragm of the diffuser disc, (ii) the net recuperation of pressure-head in the diffuser and return passages. According to the method of § 74, we find that the first item, denoted by H_c, has the value

$$H_c = \frac{(122/2)^2}{64\cdot 4}\left(1 - \left(\frac{0\cdot 22}{0\cdot 8}\right)^2\right) = 53 \text{ ft.}$$

For a pump of low-specific speed, the second item, h_g, will amount to about $0\cdot 25 \times H_{es}$, where H_{es} is the head per stage. In this case, therefore, $h_g = 0\cdot 25 \times 255 = 63$ ft.

If H' represents the head at the outlet of the impeller shown in section in Fig. 84, and the annular clearance under discussion is the one between the boss of this impeller and the neck-bush of the diffuser disc, then at the left-hand end of the clearance the head is $H' - H_c$, and at the right-hand end the head is $H' + h_g$. The net head difference is thus $53 + 63 = 116$ ft.

The method of § 83 will yield the volumetric leakage flow, thus :—

$$\begin{aligned} C_d &= \text{coefficient of discharge} = \text{about } 0\cdot 65, \\ H_l &= \text{head difference} = 116 \text{ ft.}, \\ q_l &= C_d(\pi d_c b_l)\sqrt{2gH_l}, \\ &= 0\cdot 65\left(3\cdot 14 \times 0\cdot 22 \times \frac{0\cdot 015}{12}\right)\sqrt{2 \times 32\cdot 2 \times 116}, \\ &= \mathit{0\cdot 048\ cu.\ ft./sec.} \end{aligned}$$

This amounts to 3·6 per cent. of the main flow Q.

(*Note.*—If the impellers had been set back-to-back, as in the two-stage pump in Fig. 78, then the head-difference creating leakage flow would be exactly equal to the head per stage H_{es}, and q_l would have the value 5·3 per cent. of Q.

In the 6-stage split-casing pump shown in Fig. 81, § 120, the head-difference in the first four clearance-spaces amounts to $H_{es} + H_c + h_g = 371$ ft. Thus $q_l = 6\cdot 5$ per cent. of Q.

In estimating the *power* loss associated with inter-stage leakage, Example 20, it is to be remembered that the total number of annular leakage areas is one *less* than the number of stages, viz. : number of annular spaces $= m - 1$.)

EXAMPLE 17.

To estimate the leakage loss in the hydraulic balancing system of a multi-stage pump, § 124.

Data :—

Pump as in Example 15.
Shaft sleeve diameter or impeller boss diameter = 0·22 ft.
Radial clearance between boss and neck-bush = 0·010 in.
Diameter of balance disc = 0·5 ft.

Method :—

By the method of Example 10, or § 74, it will be found that the unbalanced axial thrust on each impeller is about 1300 lb. Since

there are five impellers, the gross thrust P_a to be resisted by the balance disc amounts to $5 \times 1300 = 6500$ lb.

The available water pressure acting on the balance disc is wh_{ld}, Fig. 87.

The available area of the disc is $(\pi/4)(0{\cdot}5^2 - 0{\cdot}22^2) = 0{\cdot}157$ sq. ft.

The density of the water is 58·8 lb./cu. ft.

Since $P_g = wh_{ld} \times 0{\cdot}157$, it follows that

$$h_{ld} = 703 \text{ ft.}$$

Now h_{lc} = head-drop in annular clearance $= H_{md} - h_{ld}$
$= 1255$ (assumed) $- 703 = 552$ft.

Finally, the volumetric leakage loss is found by the method of Example 16, viz. :—

$$q_l = 0{\cdot}65 \times \left(3{\cdot}14 \times 0{\cdot}22 \times \frac{0{\cdot}10}{12}\right) \times 8{\cdot}03 \times \sqrt{522}$$
$$= 0{\cdot}070 \text{ cu. ft./sec.}$$

This quite appreciable loss, which amounts to 5·3 per cent. of the flow through the pump, might be reduced by cutting down the radial clearance between impeller boss and neckbush, Fig. 86. Only experienced design and workmanship will permit this to be done safely

EXAMPLE 18.

To assess the performance of a centrifugal pump forcing cold water through a pipe system, § 163.

Data :—

The system is generally as shown in Fig. 109. Details are :—

Total static lift	46·4 ft.
Static suction lift	6·7 ft.
Discharge	1320 gall./min.
Power input	30·5 h.p.
Total length of suction pipe	28 ft.
Diameter of suction pipe and suction branches .	10 in.
Total length of delivery pipe	156 ft.
Diameter of delivery pipe and delivery branches .	8 in.

Method :—

(i) Velocity heads may be computed thus :—

q = discharge $= 1320/374 = 3{\cdot}53$ cu. ft./sec.

v_s = velocity in suction pipe $= q/a = 6{\cdot}48$ ft./sec.

$h_{vs} = v_s^2/2g$ = velocity head in suction pipe $= 0{\cdot}65$ ft.

v_d = velocity in delivery pipe $= 10{\cdot}1$ ft./sec.

$h_{vd} = v_d^2/2g$ = velocity head in delivery pipe $= 1{\cdot}59$ ft.

(ii) Individual energy losses can now be assessed :—

h_{in} = inlet loss in foot-valve, § 290, $= 2 \times h_{vs} = 1{\cdot}30$ ft.

h_{fs} = friction loss in suction pipe $= \dfrac{4fl_s}{d_s} \cdot \dfrac{v_s^2}{2g}$.

A reasonable value for the pipe coefficient "f" is 0·006, whence $h_{fs} = \dfrac{4 \times 0{\cdot}006 \times 28}{0{\cdot}83} \times 0{\cdot}65 = 0{\cdot}52$ ft.

h_{ls} = eddy or secondary loss in suction pipe due to 2 bends $= 2 \times 0{\cdot}25 \times h_{vs} = 0{\cdot}32$ ft.

h_{fd} = friction loss in delivery pipe =

$$\frac{4 \times 0{\cdot}006 \times 156}{0{\cdot}67} \times 1{\cdot}59 = 8{\cdot}90 \text{ ft.}$$

h_{ld} = eddy loss in delivery pipe $= 0{\cdot}25 \times h_{vd} = 0{\cdot}40$ ft.

(iii) Final summation can be carried out thus :—

H_{ms} = manometric suction head $= h_s + h_{in} + h_{fs} + h_{ls} + h_{vs} = 9{\cdot}49$ ft.

H_{md} = manometric delivery head $= H_s - h_s + h_{fd} + h_{ld}$ $= 46{\cdot}4 - 6{\cdot}7 + 8{\cdot}9 + 0{\cdot}4 = 49{\cdot}0$ ft.

H_m = total manometric head $= H_{ms} + H_{md} = 58{\cdot}5$ ft.

H_e = effective head on pump $= H_m + h_{vd} - h_{vs} = 59{\cdot}4$ ft.

P_w = water horse power $= \dfrac{59{\cdot}4 \times 1320 \times 10}{33000} = 23{\cdot}8$ h.p.

η_m = gross efficiency $= P_w/P_s = 23{\cdot}8/30{\cdot}5 = 0{\cdot}78$.

EXAMPLE 19.

To estimate the power required in a boiler-feed system, § 163.

Data :—

Discharge	80,000 lb./hour
Water temperature	105° F.
Condenser vacuum	28·4 in. mercury
Boiler pressure	315 lb./sq. in.
Static head from condenser water level to boiler water level	16 ft.
Total energy loss in piping, etc.	9 ft. head

Method :—

Water density = 62·0 lb./cu. ft.
Mercury density $= 13{\cdot}6 \times 62{\cdot}4 = 846$ lb./cu. ft.
Therefore head equivalent to condenser vacuum

$$= \frac{28{\cdot}4}{12} \times \frac{846}{62{\cdot}0} = 32{\cdot}4 \text{ ft.}$$

Head equivalent to boiler pressure $= \dfrac{315}{62{\cdot}0} \times 144 = 735$ ft.

Total head to be generated by pump (or pumps)

$$= 32{\cdot}4 + 9 + 16 + 735 = 792 \text{ ft.}$$

P_w = water horse power $= \dfrac{80{,}000 \times 792}{33{,}000 \times 60} = 32{\cdot}0$ h.p.

If the pump efficiency were 72 per cent., then

$$P_s = \text{power input} = 32{\cdot}0/0{\cdot}72 = 44{\cdot}5 \text{ h.p.}$$

(*Note* that all heads are expressed in terms of the density of the water *passing through the pump.*)

EXAMPLE 20.

To compare the disc friction power loss and the leakage loss in a single-stage side-inlet pump and in a 3-stage pump, §§ 189 to 192.

Data :—

Total head	83 m.
Discharge	70 lit./sec.
Speed	1450 r.p.m.
Speed ratio	1·03

Method :—

By the usual procedure (Chaps. VII, IX), the following results are obtained :—

	Single-stage	3-*stage*
Impeller diameter, d_2 .	1·80 ft.	1·04 ft.
Shape number, n_s . .	42	97

(i) *Disc friction loss.* From equation (13-2), the following approximate values are found :—

Single-stage : $P_d = 0{\cdot}37 \times \left(\dfrac{1450}{1000}\right)^3 \times (1{\cdot}80)^5 = 21{\cdot}2$ h.p.

Multi-stage : $P_d = 0{\cdot}37 \times \left(\dfrac{1450}{1000}\right)^3 \times (1{\cdot}04)^5 = 1{\cdot}37$ h.p. per stage.

$\therefore$ Power loss *per pump* $= 1{\cdot}37 \times 3 = 4{\cdot}1$ h.p.

(ii) *Leakage power loss.* (*a*) Resulting from flow through sealing-rings, § 83.

From Fig. 54, it appears that the diameter d_c of the clearance passage will be about :—

$$d_c = 0{\cdot}76 \text{ ft. for single-stage pump}$$
$$= 0{\cdot}60 \text{ ft. for multi-stage pump.}$$

In the formula $q_l = C_d(\pi d_c b_l)\sqrt{2gH_l}$,
we may take $C_d = 0{\cdot}60$, $b_l = 0{\cdot}02$ in., $H_l = H_e$ per stage,
from which q_l = leakage flow per impeller
$= 0{\cdot}32$ cu. ft./sec. (single-stage),
$0{\cdot}145$ cu. ft./sec. (multi-stage).

Taking an efficiency of the " auxiliary " or " leakage " impeller as 0·7, then

$$P_l = \text{power loss in single-stage pump} = \frac{62{\cdot}4 \times 0{\cdot}32 \times 272}{550 \times 0{\cdot}7}$$
$$= 14{\cdot}0 \text{ h.p.}$$

P_l = power loss in three-stage pump

$$= \frac{3 \times 62{\cdot}4 \times 0{\cdot}145 \times 272}{550 \times 0{\cdot}7 \times 3} = 6{\cdot}4 \text{ h.p.}$$

(b) Resulting from other causes.

Single-stage pump. There will certainly be an additional leakage loss due to the escape of water through the balancing holes, §§ 77, 191.

Multi-stage pump. Inter-stage leakage and leakage past the balance disc will account for a substantial power loss, Examples 16, 17. In this connection the correct estimation of the head creating leakage flow is specially important. (See note at end of Example 16.) Whereas the *volumetric* leakage loss varies as $H_l^{0.5}$, the *power* loss varies as $H_l^{1.5}$.

(iii) *General comparisons* Although the above figures are purely provisional and tentative, they do show very clearly the advantages of the multi-stage pump By increasing the specific speed per impeller, we undoubtedly reduce the disc friction power loss, and we may even reduce the gross or overall leakage power loss.

EXAMPLE 21.

To assess the effect of varying the width of a centrifugal pump impeller, § 197.

Data :—

Pump " A " : Impeller diameter = 32 cm. ; impeller width = 1·5 cm. ; speed ratio = 0·98 ; flow ratio = 0·12 ; speed = 1460 r.p.m. ; gross efficiency = 0·75. The relative power losses are :—

Δ_b = Mechanical	0·03
Δ_d = Disc friction	0·09
Δ_l = Leakage	0·06

Pump " B " : Dimensions and velocities generally as in Pump " A ", except that impeller width is 3·0 cm.

Method :—

By the usual rules it is found that :—

	Pump " A "	Pump " B "
Head	31·5 m.	31·5 m.
Discharge	45·2 lit./sec.	90·4 lit./sec.
Power output . . .	19·1 h.p.	38·2 h.p.
Shape number . . .	70	99

As the total power loss in pump " A " is

$$P_s - P_w = P_w/\eta_m - P_w = 25{\cdot}5 - 19{\cdot}1 = 6{\cdot}4 \text{ h.p.},$$

we can write :—

$$P_b = 0{\cdot}03 \times 25{\cdot}5 = 0{\cdot}77,$$
$$P_d = 0{\cdot}09 \times 25{\cdot}5 = 2{\cdot}30,$$

$P_l = 0{\cdot}06 \times 25{\cdot}5 = 1{\cdot}53$,
$P_h = 6{\cdot}4 - 0{\cdot}77 - 2{\cdot}30 - 1{\cdot}53 = 1{\cdot}80$.

In pump " B " losses can be roughly estimated thus :—

$P_b = 0{\cdot}77$,
$P_d = 2{\cdot}30$,
$P_l = 1{\cdot}53$,
$P_h = 2 \times 1{\cdot}80 = 3{\cdot}60$,
$P_s = 38{\cdot}2 + 0{\cdot}77 + 2{\cdot}30 + 1{\cdot}53 + 3{\cdot}60 = 46{\cdot}40$.

Therefore efficiency $\eta_m = 38{\cdot}2/46{\cdot}40 = 0{\cdot}82$, which is a considerable improvement on the efficiency of pump " A ". It is the effects of these changes that are shown in Figs. 122, 123.

EXAMPLE 22.

To plot a complete set of pump characteristics from given test observations, § 212.

Data :—

Pump: Double-inlet centrifugal, as Fig. 44 (i), (not new). Suction and delivery branches: 8-in. diameter. Speed: held steady at 900 r.p.m. Pressure-head measured by dial type vacuum and pressure gauges, mounted level with pump axis. Discharge measured by Venturi meter, and regulated by throttle-valve. Torque measured by torque arm on swinging-yoke motor (weight of jockey-weight = 30·0 lb.). Liquid: cold water.

Method :—

The test observations are recorded on a log-sheet similar to this one :—

Speed (r.p.m.)	900		
H_{ms} = suction gauge reading (feet head)	13·41		
H_{md} = delivery gauge reading (feet head)	13·12		
Q = Venturi meter reading (galls./min.)	1265		
R = reading on torque arm = distance of jockey-weight from zero position (feet) . .	2·805		

Final results can be computed thus :—

$H_e = H_{ms} + H_{md}$ (because the pump branches are of equal diameter),

$$P_w = \frac{Q \times 10 \times H_e}{33{,}000} = \frac{Q \times H_e}{3300},$$

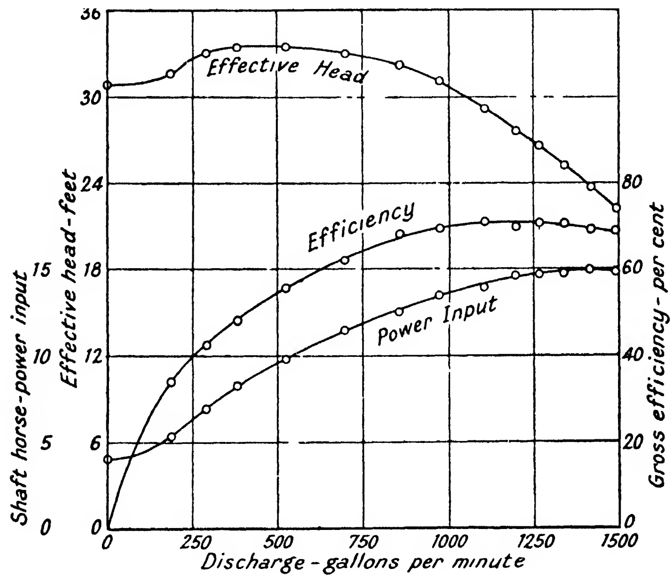
$$P_s = \frac{\text{Torque} \times \omega}{550} = \frac{30 \times R \times 2\pi \times 900}{550 \times 60} = 5{\cdot}15\,R,$$

$\eta_m = P_w/P_s$.

A suitable form of tabular statement would be

H_e = effective head (feet)	26·53			
Q = discharge (galls./min.)	1265			
P_w = output h.p.	10·15			
P_s = input h.p.	14·45			
η_m = gross efficiency	0·705			

Values are then plotted as shown in the figure.



EXAMPLE 23.

Having been given the head-discharge characteristic for one speed, to plot the head-discharge characteristic for any other speed, § 221.

Data :—

Performance of pump when running at 500 r.p.m. is given in diagram on next page. The curve for 400 r.p.m. is required.

Method :—

(i) Select three points, A, B, C, well spaced out along the given curve.

(ii) If A', B', C', are the corresponding points on the new curve, then their co-ordinates can be found thus :—

$$H_A' = H_A \times \left(\frac{400}{500}\right)^2 = 0{\cdot}64\, H_A,$$

$$Q_A' = Q_A \times \left(\frac{400}{500}\right) = 0{\cdot}8\ Q_A.$$

(iii) Having located these new points, as in the diagram, a smooth curve drawn through them will represent the desired 400 r.p.m. characteristic.

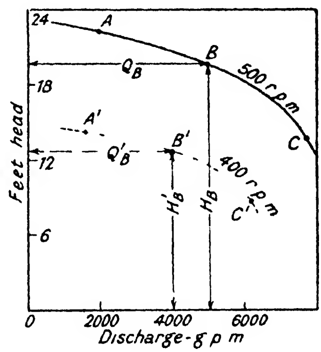
EXAMPLE 24.

Having given the head and efficiency characteristics for any one speed, to estimate the speed and power required to enable the pump to deliver a stated discharge against a stated head, § 222.

Data :—

The pump is the one whose characteristics for a speed of 900 r.p.m. are reproduced in Fig. 136. This pump is required to deliver 1700 gall/min. against a head of 135 ft.

Method :—



(i) Establish on the graph the point (a) representing the stipulated conditions (see diagram overleaf).

(ii) On the parabola that is to pass through point (a), choose some other point (b) near the given head-discharge characteristic. If we let the head H_b represented by point (b) be 80 ft., then the corresponding discharge Q_b can be found from the expression: $\left(\frac{135}{80}\right) = \left(\frac{1700}{Q_b}\right)^2$, from which $Q_b = 1310$ gall./min.

(iii) Sketch in the parabola itself, passing through points (a), (b), and the zero point of the graph.

(iv) If point (c) shows the intersection between the parabola and the original head-discharge curve, note the corresponding discharge $Q_c = 1420$ gall./min.

(v) Since, for points such as (a), (b), and (c), we know that discharge varies as speed, we can write :—

$$N_a = N_c \times \frac{Q_a}{Q_c} = 900 \times \frac{1700}{1420} = 1080 \text{ r.p.m.}$$

= required new speed.

(vi) From the given efficiency curve, note the efficiency corresponding to point (c), or discharge 1420 gall./min. This value, viz. 0·72, can be taken to apply also to point (a). Hence required power $= P_s$

$$= \frac{1700 \times 10 \times 135}{33{,}000 \times 0{\cdot}72} = 97 \text{ h.p. (approx.).}$$

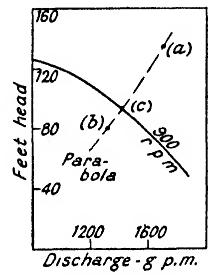
EXAMPLE 25.

To predict pump performance by means of a complete set of pump characteristics, § 222.

Data :—

The pump has the characteristics shown in Fig. 152. It is required to deliver 235 lit./sec. against a dead head of 19·8 m., and to force the water through 400 m. of 35-cm. diameter piping. What should be the speed and power consumption ?

Method—



(i) From reference charts or tables, it is found that the virtual slope in the pipe has a value of about 0·017. Therefore frictional head loss = 0·017 × 400 = 6·8 m. The secondary losses, velocity head, etc., may be estimated at 1 m., whence : total head = 19·8 + 6·8 + 1·0 = 27·6 m.

(ii) The performance point H_r = 27·6 m., Q = 235 lit./sec., can now be located on the chart, Fig. 152. By interpolation, this point is seen to correspond also to the values :—

Speed = 920 r.p.m. Efficiency = 76 per cent.

The power input will therefore be $P_s = \dfrac{235 \times 27{\cdot}6}{75 \times 0{\cdot}76}$
= 114 h.p.

EXAMPLE 26.

To predict the performance of a prototype pump from test results on a scale model, § 225.

Data :

Model	Head	171 ft.
	Discharge	5·65 cu. ft./sec.
	Speed	2000 r.p.m.
	Efficiency	0·852.
	Impeller diameter	1·0 ft.
	Scale of model	1 to 4.
	Speed of prototype pump	750 r.p.m.

Method :—

(i) From equation (15-2),

$$\eta_m = \text{efficiency of model} = 0{\cdot}852 = 1 - \frac{K}{(2000 \times 1)^{0{\cdot}2}},$$

$$\eta_M = \text{efficiency of prototype} = 1 - \frac{K}{(750 \times 16)^{0{\cdot}2}}$$

$$= 1 - 0{\cdot}148\left(\frac{2000}{750 \times 16}\right)^{0{\cdot}2} = 0{\cdot}896.$$

(ii) Applying the affinity laws of § 224, it is seen that :—

$$\text{Prototype head} = 171 \times \left(\frac{750 \times 4}{2000 \times 1}\right)^2 = 384 \text{ ft.}$$

$$\text{Prototype discharge} = 5{\cdot}65 \times \frac{750}{2000} \times \left(\frac{4}{1}\right)^3 = 135 \text{ cu. ft./sec.}$$

(iii) By the usual rules :—

$$\text{Prototype input power } P_s = \frac{135 \times 62{\cdot}4 \times 384}{550 \times 0{\cdot}896}$$
$$= 6600 \text{ h.p. (approx.).}$$

EXAMPLE 27.

To use non-dimensional characteristics for estimating pump performance, § 228.

Data :—

Type of pump	Mixed-flow centrifugal.
Head	32 ft.
Speed	780 r.p.m.
Rotor diameter	1·4 ft.
Shape number at design point . .	210.

What discharge would this pump give ?

Method :—

(i) Compute the characteristic head number h_c =

$$\frac{gH}{n^2D^2} = \frac{32{\cdot}2 \times 32}{\left(\frac{780}{60}\right)^2 \times (1{\cdot}4)^2} = 3{\cdot}08.$$

(ii) Selecting from Fig. 155 the characteristic curve for $n_s = 210$, we find that the value of q_c corresponding to $h_c = 3{\cdot}08$ is 0·33.

(iii) Inserting appropriate values in the expression for discharge number gives the result :—

$$q_c = 0{\cdot}33 = \frac{q}{nD^3} = \frac{q}{\left(\frac{780}{60}\right) \times (1{\cdot}4)^3},$$

or q = discharge of pump = 12 cu. ft./sec. (approx.).

Manifestly this is only a tentative value, but from the general run of the curves it seems likely that the figure cannot be more than a few per cent. in error.

EXAMPLE 28.

To estimate the performance of a pump when pumping petroleum products, § 233.

Data :—

A single-stage centrifugal pump when working with water gave the following performance :—

Head = 32·6 m.; discharge = 40·5 lit./sec.; speed = 1460 r.p.m.; efficiency = 75 per cent.; impeller diameter = 36 cm.

The petroleum products are :—

(*a*) *Kerosene*, specific gravity = 0·82,
viscosity = 30 sec. Redwood.

(*b*) *Fuel oil*, specific gravity = 0·94,
viscosity = 920 sec. Redwood.

What would be the performance with these liquids, with the pump running at the same speed (1460 r.p.m.) and giving the same discharge (40·5 lit./sec.) ?

Method :—

(*a*) The figures show that the kerosene has a kinematic viscosity of about 0·02 Stokes. As this so closely resembles the viscosity of water, it can be said at once that the pump efficiency will not be appreciably affected. Therefore

$$H_e = \text{head} = 32{\cdot}6 \text{ m.}$$

and

$$P_s = \text{power input} = \frac{40{\cdot}5 \times 0{\cdot}82 \times 32{\cdot}6}{75 \times 0{\cdot}75}$$

$$= 19{\cdot}3 \text{ h.p.}$$

(*b*) The kinematic viscosity of the fuel oil is about 2·4 stokes. Using this value, we find from Fig. 158 that for the given impeller size the *head* is about 86 per cent. of the equivalent head for water, while the *efficiency* is about 76 per cent. of the "water" efficiency.

$$\text{Thus } \textit{head of oil} = 32{\cdot}6 \times 0{\cdot}86 = 28{\cdot}0 \text{ m.}$$

$$\textit{efficiency} = 0{\cdot}75 \times 0{\cdot}76 = 0{\cdot}57$$

$$\textit{power input} = \frac{40{\cdot}5 \times 0{\cdot}94 \times 28{\cdot}0}{75 \times 0{\cdot}57} = 25{\cdot}0 \text{ h.p.}$$

It is clearly to be understood that these figures are subject to a tolerance of several per cent.

EXAMPLE 29.

To examine the possibilities of testing a pump with air, § 236.

Data :—

Estimated performance of pump working with water :—

Head	80 ft.
Speed	500 r.p.m.
Discharge	50 cu. ft./sec.

Method :—

As this is quite a fair-sized machine, absorbing at least 500 h.p., it will certainly be worth while to run preliminary tests with air, if this is at all possible.

(*a*) *Actual pump with atmospheric air.* Assuming normal density to be 0·075 lb./cu. ft., we find that the pressure generated by the

pump running at its normal speed of 500 r.p.m. would be equivalent to a water column of

$$80 \times \frac{0{\cdot}075}{62{\cdot}4} = 0{\cdot}096 \text{ ft.} = 1{\cdot}15 \text{ in.}$$

This is too small a pressure-difference to be effectively utilised under test-bed conditions; for it is all we have available for measuring not only the "head" produced by the pump but also the discharge. (Some type of flow-nozzle must necessarily be used for flow measurement.)

If it would be safe or practicable to speed up the pump to—say—1000 r.p.m., then the predicted pressure-difference would be about 4·6 in. of water, which could quite accurately be gauged.

(*b*) *Model pump with atmospheric air.* As the prototype rotor would be about 3 ft. diameter, a model to a scale of $\frac{1}{3}$ might be suitable. If it were run at 3000 r.p.m, it would generate a pressure-difference of 4·6 in of water, and the whole test-rig should therefore be capable of giving most useful information.

(*c*) *Model pump with compressed air.* Taking a working pressure in the closed circuit of 100 lb./sq in. absolute, the corresponding air density at normal temperature is 0·52 lb./cu. ft., and kinematic viscosity is $0{\cdot}21 \times 10^{-4}$ sq. ft./sec. Using the values

$$N = 3000 \text{ r.p.m.},\ v_2 = 157 \text{ ft./sec.},\ d_2 = 1 \text{ ft.},$$

it appears that the value of the Reynolds number would be :—

$$R_n = \frac{v_2 d_2}{\nu} = \frac{157 \times 1 \times 10{,}000}{0{\cdot}21} = 7{,}500{,}000.$$

Estimated performance figures are :—

Head of air $= 80 \times 4 = 320$ ft.

$$\text{Equivalent head of water} = 320 \times \frac{0{\cdot}52}{62{\cdot}4} = 2{\cdot}67 \text{ ft.}$$

$$\text{Discharge} = 50 \times \left(\frac{3000}{500}\right) \times \left(\frac{1}{27}\right) = 11{\cdot}1 \text{ cu. ft./sec.}$$

$$P_w = \text{output power} = \frac{11{\cdot}1 \times 0{\cdot}52 \times 320}{550} = 3{\cdot}36 \text{ h.p.}$$

$$P_s = \text{input power} = \text{about } 3{\cdot}8 \text{ h.p.}$$

Here also are values capable of reliable measurement. To assess the worth of the figure representing efficiency that finally emerges—the figure that we are chiefly interested in—it will be necessary to compute the value of the Reynolds number of the prototype pump running normally on water. The relative data are: $v_2 = 78{\cdot}5$ ft./sec., $d_2 = 3$ ft., $\nu = 0{\cdot}0000108$ sq. ft./sec., from which

$$R_n = \frac{78{\cdot}5 \times 3}{0{\cdot}0000108} = 22{,}000{,}000.$$

Evidently, then, the "model" Reynolds number is still substantially below the prototype number, and hence the efficiency of the prototype pump may reasonably be expected to be more favourable than the measured efficiency of the model.

EXAMPLE 30.

To compare the behaviour of constant-speed and variable-speed pumps when working against a constant head at varying discharge, § 242.

Data :—

The pump has the characteristics shown in Fig. 152.

Constant head = 25.0 m.
Maximum speed = 980 r.p.m.
Maximum discharge at this speed = 288 lit./sec.
Reduced discharge = 120 lit./sec.

Pumping set " A " is driven by a constant-speed motor running at 980 r.p.m., and discharge is regulated by throttling.

Pump " B " is driven by a variable-speed motor.

Method :—

Pump " *A* ". From Fig. 152, we find that at a speed of 980 r.p.m. the values corresponding to $Q = 120$ lit./sec. are :—

Head = 37·5 m. Efficiency = 0·73.

Hence P_s = power input to pump = B.H.P. of motor

$$= \frac{120 \times 37{\cdot}5}{75 \times 0{\cdot}73} = 82{\cdot}2 \text{ h.p.}$$

The power wasted in the throttle-valve

$$= \frac{(37{\cdot}5 - 25) \times 120}{75} = 20{\cdot}0 \text{ h.p.},$$

which calls for a useless motor B.H.P. of $20{\cdot}0/0{\cdot}73 = 27{\cdot}4$ h.p.

Pump " *B* ". The pump is here slowed down until it delivers just the stipulated quantity. The corresponding values taken from Fig. 152 are then :—

Speed = 805 r.p.m. Discharge = 120 lit./sec.
Head = 25 m. Efficiency = 0·75.

Hence the motor B.H.P. is now $\dfrac{120 \times 25}{75 \times 0{\cdot}75} = 53.4.$

(*Note* that the particular data here chosen tend to exaggerate the defects of the constant-speed system. If the pump characteristic had been such that its design point corresponded with the maximum discharge of 288 lit./sec., the power wasted at reduced discharge would have been much less.)

EXAMPLE 31.

To estimate the time required by a constant-speed pump to lift water from a lower tank to an upper one, § 241.

Data :—

Pump : At design point, head = 50 ft.; discharge = 400 g.p.m. Head-discharge characteristic as at (i), Fig. 144.

Tanks : Each rectangular, 30 ft. long × 15 ft. wide.

Difference in water level when upper tank is empty and pumping begins is 39 ft.

Total quantity to be lifted = 42,000 gallons.
(Inlet to upper tank is always submerged.)

Method :—

(i) As the lower tank is progressively emptied, and the upper tank filled, the head on the pump will steadily increase and the rate of discharge will decline. The total time of pumping can be divided into a number of short periods—say 4—during each of which the mean discharge may be assumed to remain unchanged.

If t (min.) is the duration of the period,
Q gallons per minute is the mean discharge,
h feet is the change in head,
A is the surface area of each tank = 450 sq. ft.,

then $Q \times t = A \times h/2$, or $t = \dfrac{450 \times 6{\cdot}23 \times h}{Q \times 2} = \dfrac{1400h}{Q}$.

(ii) Total variation of level in one tank

$$= \frac{42{,}000}{6{\cdot}23 \times 450} = 15 \text{ ft.}$$

Therefore total variation of head on pump = 30 ft.
From pump characteristic, it is found that :—
At start of pumping, head = 39 ft., discharge = 500 gall./min.
At end of pumping, head = 69 ft., discharge = 100 gall./min.
The duration of each period, t, may be that in which the pump discharge changes by 100 gall./min. Thus the mean discharge in each of the respective periods will be 450, 350, 250, and 150 gall./min.

(iii) By scaling off the change of head h corresponding with the change of discharge 100, the durations of the periods and the required total time are found thus :—

Mean Q .	450	350	250	150	
h (feet) .	10·8	8·8	6·7	3·7	(Total = 30)
$t = \dfrac{1400\,h}{Q}$.	33·7	35·3	37·5	34·5	(Total = 141 minutes)

(*Note*.—The water can be imagined to return from the upper to the lower tank through appliances of one sort or another that require variable rates of flow. When this return flow begins, the whole circuit is in principle automatically self-regulating. The greater the demand for water, the lower the level in the upper tank falls, and the greater the discharge from the pump becomes. If no water at all is needed, the upper tank level will rise until the pump discharge ceases. The pump could still be left running without harm, provided that a leak-off device were fitted, § 206, and that priming troubles were eliminated by giving the pump a positive inlet pressure, § 291.)

EXAMPLE 32.

To examine the performance of pumps in parallel when forcing water through a long pipe, § 243.

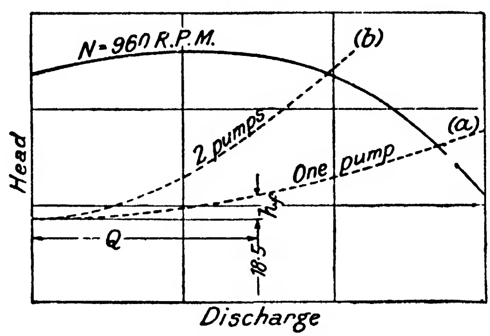
Data :—

Pumps : Constant-speed, 960 r.p.m., characteristics as Fig. 152.
Static head = 18·5 m.
Pipe : Length = 2180 m. ; diameter = 0·5 m.

Method :—

(i) On the graph showing the individual pump performance, the external head characteristics must be plotted as in the diagram. The tabular statement shows the procedure :—

Discharge in Pipe, and Discharge per Pump when one Pump working (lit./sec.).	Discharge per Pump when two Pumps working (lit./sec.)	Virtual Slope i (from Discharge Charts).	Friction loss in pipe, $h_f = 2180 \cdot i$ (metres).
50	—	0·00014	0·3
100	50	0·0005	1·1
150	-	0·0012	2·6
200	100	0·002	4·4
250	—	0·003	6·5
etc.	—		etc.



For present purposes, it can be assumed that H_e = effective head = static head H_s + friction head h_f.

(ii) From the intersection of the external head characteristics, (*a*) and (*b*), with the pump characteristic, the desired information can at once be read off :—

(*a*) *One pump working* : $Q = 268$ lit./sec.
$H_e = 26·1$ m.
$\eta_m = 0·72$

from which : motor output = 130 b.h.p.

(*b*) *Two pumps working* : $Q = 195$ *lit./sec.* per pump

$$H_e = 33{\cdot}8 \text{ m.}$$
$$\eta_m = 0{\cdot}79$$
$$\text{Motor output} = P_s = 111 \text{ h.p. per pump.}$$

(*Note* that switching in the second pump only increases the total discharge by 46 per cent., but increases the power demand by 70 per cent.)

EXAMPLE 33.

To estimate the limiting static suction lift for a given installation, § 253.

Data :—

The pump and piping are as in Example 18. Pump speed = 980 r.p.m. ; pump type, double suction ; altitude of installation, 2850 ft. above sea-level ; temperature of water, 125° F. ; $H_e = 59{\cdot}4$ ft.

Method :—

(i) The data can be transformed as follows :—

p_a = atmospheric pressure at given altitude = 13·2 lb./sq. in.
w = density of water at given temperature = 61·6 lb./cu. ft.
p_{vp} = vapour pressure at given temperature = 2·0 lb./sq. in.
N_s = specific speed of pump = 29 (foot).

(ii) The values of the basic terms can then be computed :—

σ = "sigma" = cavitation factor = 0·10, from Fig. 180,

$$H_{int} = \sigma H_e = 0{\cdot}10 \times 59{\cdot}4 = 5{\cdot}94 \text{ ft.}$$

$$H_t = \frac{p_a - p_{vp}}{w} = \frac{(13{\cdot}2 - 2{\cdot}0) \times 144}{61{\cdot}6} = 26{\cdot}2 \text{ ft.},$$

$$H_{ms} = H_t - H_{int} = 26{\cdot}2 - 5{\cdot}94 = 20{\cdot}3 \text{ ft.}$$

= manometric suction head,

h_{sc} = limiting static suction head = $H_{ms} - h_{in} - h_{fls} - h_{vs}$
$= 20{\cdot}3 - 1{\cdot}30 - 0{\cdot}84 - 0{\cdot}65,$
$= 17{\cdot}5$ ft. (approx.).

EXAMPLE 34.

To compare starting conditions for (*a*) a centrifugal pump, (*b*) a propeller pump, when forcing water through a long pipe, § 266.

Data :—

Pumps : Head at design point = 20 ft. Discharge at design point = 10 cu. ft./sec. Characteristics as in diagram ; constant speed. Static head = 8 ft. Pipe : 5460 ft. long ; 2 ft. diameter. Friction loss is 12 ft. when discharge is 10 cu. ft./sec.

Method :—

(i) As we are here concerned only with the mean rate of acceleration of the water column in the pipe, the pressure-waves can be disregarded. Time may be reckoned from the moment at which flow first begins, without allowing for the period required to bring the rotating parts up to full speed.

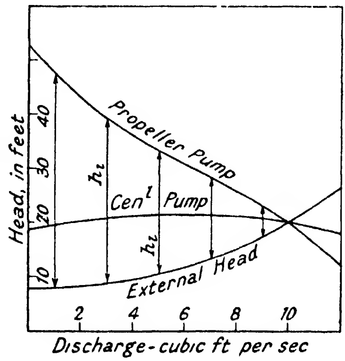
(ii) The external head curve is first plotted, as in Example 32. Then at any moment the accelerating head h_i is the difference between the internal head and the external head, viz., $h_i = H_e - H_s - H_f$. This quantity can be scaled from the diagram.

(iii) The total time of acceleration can be divided into (say) five periods, each one corresponding to a change of discharge of 2 cu. ft./sec., viz., a change of velocity of 0·64 ft./sec. If a is the mean rate of acceleration during such a period, of duration t secs., then

$$0{\cdot}64 = a.t.$$

But, from § 262,

$$a = \frac{gh_i}{l} = \frac{32{\cdot}2 \times h_i}{5460}.$$



Consequently $t = \dfrac{0{\cdot}64 \times 5460}{32{\cdot}2 \times h_i} = \dfrac{109}{h_i}$.

(iv) By inserting in this expression values of h_i scaled from the diagram, we obtain the following values for t during successive periods :—

For centrifugal pump : 9·4 : 9·4 : 10·6 : 14·4 : 34·0 sec.
Total = 77·8 sec.

For propeller pump : 2·7 : 3·5 : 4·7 : 7·3 : 18·5 sec.
Total = 36·7 sec.

These figures mean that after about 78 sec. (or 37 sec., as the case may be), the liquid column in the pipe has settled down to its terminal velocity and the whole installation is running steadily. Although the propeller pump accelerates the water much more rapidly, yet in other respects the installation is not well off. Both the head at starting, and the power at starting, are much greater in the propeller pump installation than they are in the centrifugal pump installation. See §§ 213, 280, 327.

EXAMPLE 35.

To estimate the value of the surge pressure in a motor-driven pumping installation, § 269.

Data :—

Pumps : two sets, each thus :—

Normal speed	1200 r.p.m.
Normal head	284 ft.
Normal discharge	7850 gall./min.
Moment of inertia of revolving parts	1450 lb. ft.^{2}

Pipe :—
Length . . . . 900 ft.
Diameter . . . . . . 3·5 ft.
Pipe friction assumed negligible.

Method :

(i) The velocity v_0 of the pressure-wave along the pipe may be taken as 4000 ft./sec. Hence time dt for wave to make one journey along pipe $= l/v_0 = 900/4000 = 0{\cdot}225$ sec.

(ii) Applying formula (17-3) to the present system, we find :—
dN = drop in speed of revolving elements of pump and motor in time dt sec., while pressure-wave makes one trip along pipe,

$$= \frac{dt \, . \, K_t \, . \, W \, . \, H_e}{\eta_m N I},$$

$$= \frac{0{\cdot}225 \times 2950 \times WH_e}{\eta_m N \times 1450} = 0{\cdot}46 \, \frac{WH_e}{\eta_m N}.$$

(iii) The initial values to be inserted in this expression are :—

$$W = \frac{7850 \times 10}{60} = 1300 \text{ lb./sec. per pump},$$

H_e = head = 284 ft.
η_m = pump efficiency = 0·85 (assumed),
N = pump speed = 1200 r.p.m.,

from which dN = drop in speed during first period = 167 r.p.m. These values can now all be put in tabular form thus :—

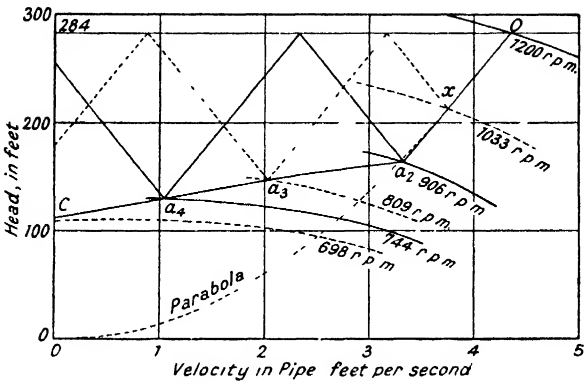
No. of Period.	Conditions at Beginning of Period.				Drop in Speed, dN.
	W.	H_e.	η_m.	N.	
1	1300	284	0·85	1200	167
2	1130	214	0·85	1033	127

(iv) In the $v - h$ diagram, Fig. 190, initial value of v = velocity in pipe $= \dfrac{2 \times 1300}{62{\cdot}4 \times 0{\cdot}7854 \times (3{\cdot}5)^2} = 4{\cdot}35$ ft./sec.

Initial value of head = 284 ft. (neglecting friction). This point O can be plotted on the graph (see figure overleaf), and the head-velocity characteristic curve sketched in, for the initial speed of 1200 r.p.m.

Now the speed at the end of the first period is evidently 1200 − 167 = 1033 r.p.m. ; and the corresponding

characteristic curve can be located by the method of Example 23. The parabola of maximum efficiency is drawn through the initial point O, and a point x on this curve is chosen such that $\frac{v_x}{4\cdot35} = \frac{1033}{1200}$, or $v_x = 3\cdot77$ ft./sec. The 1033 r.p.m. characteristic is now sketched in, passing through the point x.



(v) The slope of the construction lines, $\frac{v_o}{g}$, has the value

$$\left(\frac{4000}{32\cdot2}\right) = 124 \text{ sec.}$$

The first construction line, Oa, can thus be drawn, such that point a is the intersection between this line and the 1033 r.p.m. characteristic. In the diagram, point a happens to coincide with point x. It is this point which provides the values applicable to the second period, viz., $W = 1130$, $H_e = 214$, $\eta_m = 0\cdot85$, $N = 1033$. Inserting these in turn in the expression for dN, we find that the drop in speed during the second period is 127 r.p.m.

(vi) The filling up of the tabular statement, and the plotting of the points on the diagram, proceed simultaneously, until the graph is completed. From the graph we read off :—

Minimum head at pumps = 115 ft.

Time for flow through pumps to cease, from moment of tripping out motors = 4·6 periods — about 1·0 sec.

Negative surge = 284 — 115 = about 170 ft. head.

Positive surge = 170 ft. head.

Maximum head, immediately after closing of reflux valves, = 284 + 170 = 454 ft. head, *exclusive* of slam-pressure, if any.

Pump speed at moment of closing of reflux valves = about 700 r.p.m. in *forward* direction.

(*Note.*—In the above method, speed changes corresponding to time intervals of dt have been computed, whereas in § 269 the intervals are of duration $2\,dt$. This is preferable when the speed variations are relatively great, as they are here; because the fundamental formula 17·3 is only strictly true for *very small* speed changes. Moreover, the greater the interval, the greater becomes the discrepancy between conditions at the beginning of the period, and the true conditions at the middle of the period (end of § 270). But all such errors are on the *safe* side: they tend to exaggerate the surge pressure.)

EXAMPLE 36.

To examine the advantages of using a hydraulic coupling for a pump driven by a constant-speed motor, § 284.

Data :—

As in Example 30, pump " A ", except that a hydraulic coupling is now interposed between the motor and the pump.

Method :—

In order to obtain the reduced discharge of 120 lit./sec. against the steady head of 25 m., oil is drained from the coupling until the pump speed falls to the minimum value necessary, viz., 805 r.p.m. The throttle valve is left *fully open*.

The equivalent input *to the pump* has already been found to be 53·4 h.p.

Since

$$\frac{\text{Coupling power output}}{\text{Coupling power input}} = \frac{\text{output speed}}{\text{input speed}} = \frac{\text{pump speed}}{\text{motor speed}} = \frac{805}{980};$$

$$\text{evidently motor output} = 53{\cdot}4 \times \frac{980}{805} = 65{\cdot}0 \text{ h.p.}$$

When running direct-coupled *to the* pump, the output of the constant-speed motor was seen to be 82·2 h.p., hence the saving in power realised by the coupling is 82·2 − 65·0 = 17·2 h.p.

EXAMPLE 37.

To draw up comparative preliminary schemes for a low-head pumping installation, § 304.

Data :

Effective head = 20 ft.

Total discharge = 330 cu. ft./sec.

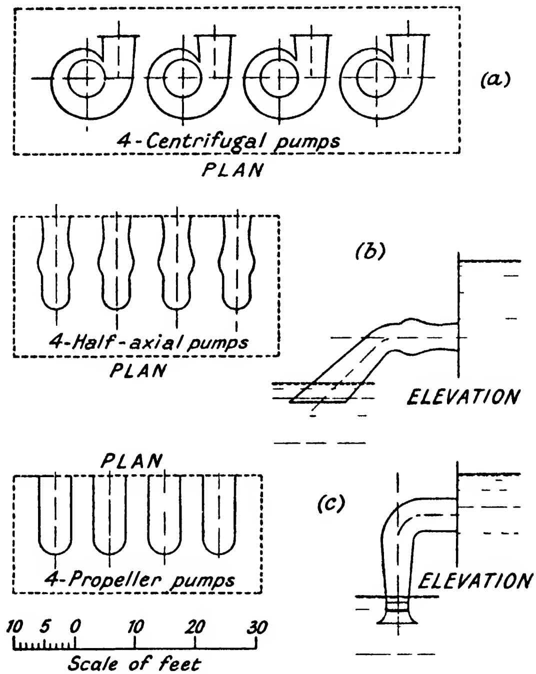
It is proposed to have three units, with an additional standby unit, all of the same size.

Centrifugal, half-axial, and propeller pumps are to be compared.

Method :—

(*a*) *Centrifugal*. A high specific-speed, side-inlet pump might be suitable, as in Fig. 61. Taking a shape number of about 200, it appears that the pump *speed* should be about 150 r.p.m., and the rotor *diameter* should be about 5·7 ft. Using the proportions of Fig. 61, we find that the outside diameter of the casing will be about 13 ft.

By the method of § 253, the maximum *static suction lift* h_{sc} is found to be about 20 ft.



(*b*) *Half-axial*. Using Fig. 66 as a guide, and choosing a shape number of 330, we find :—

Speed = 240 r.p.m.
Rotor inlet diameter = 3·4 ft.
Casing outside diameter = 5·6 ft.
Static suction lift = 13 ft.

(*c*) *Propeller*. With a shape number of 660, values are :—

Speed = 480 r.p.m.
Rotor diameter = 3·2 ft.
Casing diameter = 4 ft.
Static suction lift = — 2 ft.

Various combinations of pump, motive-unit, and transmission can now be studied, thus :—

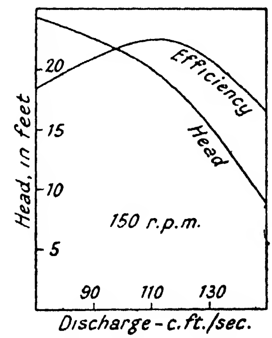
(*a*) The centrifugal pumps can most conveniently be set vertically, as in Fig. 205 or Fig. 206. A speed-reduction gear-box will certainly be needed, no matter whether the pumps are driven by oil engines or by electric motors.

(*b*) There is no option but to set the half-axial pumps as in Fig. 66. Remembering that the b.h.p. output of the motive-units will be of the order of 300 h.p. each, the shaft speed of 240 r.p.m. is not too low to permit of direct-coupling a horizontal oil engine to each pump, as in Fig. 202. A medium-speed vertical oil engine running at about 500 r.p.m. could also be geared to the pump, but what is saved in space might be lost in durability.

Electric motors would be coupled through step-down gear-boxes.

There should be little difficulty in setting the pumps low enough to meet the conditions of limiting suction lift.

(*c*) Suction conditions exert the overriding influence in the propeller pump installation. The pump must be set at least 2 ft. below inlet water level, and thus it must be installed vertically, as in Fig. 68. It seems unlikely that oil engines would fit in very well, and thus the choice lies between direct-coupled vertical-shaft motors, and geared vertical-shaft motors.



If the stage has not yet been reached at which any of these possibilities need be rejected, outline sketch plans can be prepared (see opposite page). These will give some sort of preliminary notion of the relative cost of buildings, foundations, etc.; and they will also indicate whether the conditions on site are likely to favour any particular scheme. At this point, too, the effect of varying head (if any) might be examined, as in Example 38.

As enough information will now be available to provide a fairly close estimate of power costs, it should finally be possible to pick out the most promising schemes and to prepare detailed specifications for circulation to pump makers.

EXAMPLE 38.

To examine the effect of varying head upon different systems of low-head pumping plant, § 309.

Data :—

Centrifugal pumps as in Example 37 (*a*). Performance at design point :—

Head	20 ft.
Discharge	110 cu. ft./sec.
Speed	150 r.p.m.
Efficiency	0·90

Characteristics as in diagram on previous page.

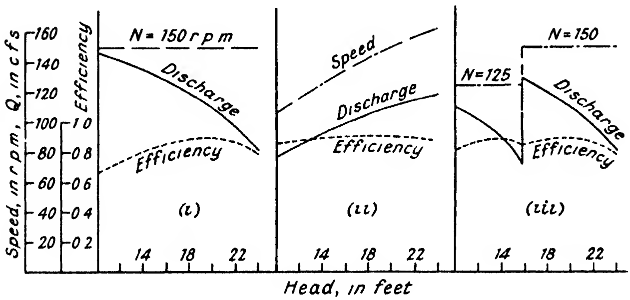
The systems to be compared are :—

(i) Constant-speed pumps.

(ii) Variable-speed pumps, regulated to maintain highest possible efficiency.

(iii) Pumps having a choice of two speeds, driven either by 2-speed motor or by constant-speed motor with 2-speed gear-box. The speed ratio is 6 : 5, whence :—

High speed = 150 r.p.m.
Low speed = 125 r.p.m.



Method :—

The most useful sort of comparison is to plot discharge, speed, and efficiency directly against head, as in the diagram. For system (i) the information can be taken directly from the given characteristics. For system (ii) the procedure of Examples 23 or 24 may be helpful.

In trying to assess the advantages of system (ii) and (iii), it will be necessary to know what is the general nature of the head fluctuation. If the pumps usually run near their designed conditions, and the head only occasionally reaches its upper and lower limits, then the additional cost and complication of the change-speed mechanism will hardly be justified. Again, if conditions of maximum *head* coincide with conditions of maximum *discharge*, system (ii) seems preferable ; but if, on the other hand, high discharge and low lift occur together, system (i) would automatically provide this type of variation. Naturally such considerations have an important bearing on the maximum power demand of the installation.

EXAMPLE 39.

To contrive the most effective pumping plant from a given used (second-hand) pump and a given engine.

Data :—

It is required to deliver the maximum possible discharge against a total head of 65 ft.

Pump. When first inspected the pump looks old, rusty, and neglected; after being stripped and cleaned it can be described thus :—

Inlet and outlet branches : 8-in. diameter.
Impeller : Double-inlet, $12\frac{1}{2}$-in. diameter, $1\frac{5}{8}$-in. wide at mouth.
Blade angle : about 20°.
Blades look pitted and sealing-rings worn.

Engine. Horizontal slow-speed, in fairly good condition. Known to be good for 55 b.h.p. at 300 r.p.m.

Method :—

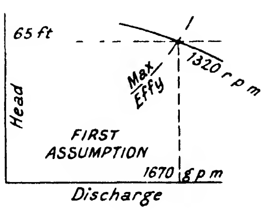
(i) As a first rough estimate, we can take the pump overall efficiency (engine b.h.p. to w.h.p.) as 0·6. Hence

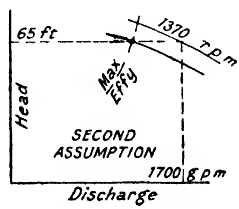
$$P_w = 0\cdot6 \times 55 = 33 = \frac{Q \times 10 \times 65}{33{,}000},$$

or Q = discharge = 1670 gall./min.
Assuming the pump to be working at its design point, a suitable value of the speed ratio ϕ would be about 1·1. Therefore

$$v_2 = 1\cdot1\sqrt{2g \times 65} = \frac{3\cdot14 \times 1\cdot04 \times N}{60},$$ from which N = speed = 1320 r.p.m., and N_s = specific speed = 41 (foot). (See diagram " First Assumption ").

(ii) The next question is, do these figures really represent the pump design conditions ? Evidence on this point can be collected thus : Using the methods of §§ 90 to 93, the outlet blade angle should be about 25°. But the measured angle is only 20°. Again, the flow ratio ψ works out at 0·16, which is rather high for the assumed values of specific speed and for the width ratio of 0·13. The specific speed itself is perhaps rather high for a pump of this size and class. On the whole, then, we are justified in believing that the discharge of 1670 gall./min. is greater than the designed dis-





charge for the given head and speed. In other words, the pump will be working under "increased-flow" conditions,

(iii) In order to allow for this, the pump speed should be slightly higher than was originally suggested—about 1370 r.p.m. would be a reasonable figure. As the original estimate of

efficiency—60 per cent.—was quite conservative, the pump should deliver at least 1700 gall./min.
(See diagram " Second Assumption ".)

(iv) To transmit the necessary power, there might be on the engine shaft a pulley 4·7 ft. by 9 in., and on the pump shaft a pulley 12-in. diameter by 9 in. An additional bearing might be necessary to ensure that the pump pulley is properly supported.

EXAMPLE 40.

To determine the type and speed of pump working in a siphonic circuit, §§ 254, 310.

Data :—

Pump to be arranged horizontally, generally as Fig. 204 (I).
Static lift = 10 ft.
Pump axis is set 5 ft. above water level in outlet channel.
Discharge = 120 ton/min.

Method :—

(i) The limiting factor here is evidently the static suction lift, which is greater than the total static lift.
For preliminary purposes we may insert in equation (16-1) the values : $h_{sc} = 15$ ft. ; $H_t = 33$ ft. ; $H_e = 10$;
$H_{ext} = \dfrac{Y^2}{2g} = \dfrac{(\psi\sqrt{2gH_e})^2}{2g}$; $H_{int} = \sigma \times 10$. This yields :—

$$15 = 33 - 10\psi^2 - 10\sigma, \text{ from which } \psi^2 + \sigma = 1{\cdot}8$$

(ii) Suitable values of σ can best be found by trial and error, thus :—

Assume specific speed $N_s = 100$ (foot),
then $\psi =$ about 0·3,
$\sigma =$ about 0·75,
$\psi^2 + \sigma = 0{\cdot}84$, which is too low.
Assume $N_s = 150$ (foot),
then $\psi =$ about 0·5,
$\sigma =$ about 1·2,
$\psi^2 + \sigma =$ about 1·45.

Remembering now that friction in the inlet pipe, and velocity head rejected at outlet, have still to be accounted for, there would be no advantage in going beyond this specific speed of 150.

(iii) From the given data, the equivalent rotational speed would be 300 r.p.m. This would be suitable either for a direct-coupled horizontal oil engine, or for a geared electric motor. In any event the given specific speed represents a *propeller pump*. Its main dimensions could be estimated as in Ex. 14.

EXAMPLE 41.

To establish the main particulars of a bore-hole pumping installation, §§ 137, 313.

Data :—

Total head on installation = 400 ft.
Depth from ground level to water level = 220 ft.
Discharge = 1000 gall./min.
Diameter of borehole = 24 in.

Method :—

(i) Shape number of borehole pump may provisionally be taken as $n_s = 100$.

$$\text{Now } Q = 1000/374 = 2{\cdot}68 \text{ cu. ft./sec.}$$
$$= 0{\cdot}0003 \times N \times (2)^3 \quad \text{(from § 137)}$$

from which N = shaft speed = 1120 r.p.m.

(ii) From the specific speed formula

$$N_s = 100 \times 0{\cdot}273 = \frac{0{\cdot}0174 \times 1120 \times \sqrt{1000}}{(H_{es})^{3/4}},$$

it appears that H_{es} = head per stage = 64 ft.
If the borehole pump were given *four* stages, the head generated would be $4 \times 64 = 256$ ft., which would give a reasonable margin over the dead lift to ground level of 220 ft.
Assuming a speed ratio of $\phi = 1{\cdot}0$, the impeller diameter works out at 1·1 ft., and the casing diameter at about 1·75 ft., § 125. Since the borehole diameter is 2·0 ft., the figures check quite well.

(iii) By the usual rules, it is found that the borehole pump output is 78 h.p., the efficiency about 0·73, and the power input is therefore 107 h.p.
Assuming a transmission-shaft diameter of 2½ in., and a length of 220 ft., the formula of § 316 shows that the transmission loss P_{sh} may be about 8 h.p.

(iv) Allowing a diameter of 10 in. for the vertical pipe between pump and ground level, the frictional loss in a plain pipe would be about 2 ft.; but because of the disturbances to flow created by the spiders, shaft-couplings, etc., it would be well to allow for a loss of 10 ft. head.

(v) The head to be generated by the booster-pump or force-pump can now be reckoned. It will amount to

$$400 + 10 - 256 = 154 \text{ ft.}$$

A 2-stage pump would serve, running at the same speed as the borehole pump, and absorbing 64 h.p. The corresponding shape number—87—would be permissible.
Thus the total power to be delivered by the motive-unit would be $107 + 8 + 64 = 179$ h.p.

(vi) The selected shaft speed—1120 r.p.m.—would be suitable for a variable-speed D.C. or A.C. motor. If a constant speed of 970 r.p.m. were specified, the borehole pump would probably require five stages instead of four.

EXAMPLE 42.

To estimate the main particulars of a high-head pumping-set with auxiliary low-lift pump, § 321.

Data :—

Total head = 720 ft.
Discharge = 5200 gall./min.
Speed = 1460 r.p.m.

Method :

(i) If the whole work were to be done by a 2-stage pump of specific speed per stage of 22 (foot), having single-inlet impellers, then the values to be inserted in formula (16-1) would be about :—

$$H_t = 33 \text{ ft.} : \ H_{ext} = 5 \text{ ft.} : \ H_e = 360 \text{ ft.} : \ \sigma = 0{\cdot}2,$$

whence h_{sc} = limiting suction static head

$$= 33 - 5 - (0{\cdot}2 \times 360) = -44 \text{ ft.}$$

This means that a positive pressure should prevail at the pump inlet flange : the pump must be set 44 ft. *below* the water surface in the suction well. This figure could be reduced if double-inlet impellers were used instead of single-inlet impellers, § 253 ; but a more effective method would be to instal an auxiliary pump that would lift the water into the main pump.

(ii) The duty of the auxiliary pump might be :—

Effective head = 50 ft.
Discharge = same as main pump = 5200 gall./min.
Type : double-suction ; specific speed = 34 (foot).

In the ordinary way it is found that the speed N should be about 500 r.p.m., and that the pump could be set at least 15 ft. *above* suction water surface.

(iii) To check now the performance of the 2-stage main pump, we note that each of its stages now generates a head of only $\frac{1}{2}(720 - 50) = 335$ ft. The internal head drop will be about $0{\cdot}2 \times 335 = 67$ ft., corresponding to a pressure drop of 29 lb./sq. in. If we allow a minimum absolute pressure in the pump passages of 1 lb./sq. in., it follows that at the main pump first-stage inlet flange, or the auxiliary pump delivery flange, the absolute pressure should be 30 lb./sq. in. On the other hand, if the pumping set is installed 15 ft. above suction water level, then the pressure-head at the auxiliary pump delivery flange will be $50 - 15 = 35$ ft., and the

corresponding absolute pressure is $(35 \times 0{\cdot}434) + 14{\cdot}7 =$ 30 lb./sq. in. The two values thus check very well.

(iv) Remembering that the maximum value of the cavitation factor σ has been chosen, it looks safe to accept the system as it is, without making further allowance. But if there were any doubt, the set might be lowered another 5 ft., bringing the suction lift down to 10 ft. The whole pumping set would then correspond exactly with Fig. 213 (*a*), and could be designed to fulfil the conditions :—

Main pump head = 670 ft.
Main pump speed = 1460 r.p.m.
Auxiliary pump head = 50 ft.
Auxiliary pump speed = 500 r.p.m.
Step-down gear-box ratio = 3 : 1 (about).

EXAMPLE 43.

To plan the pumping installations in a chain of pumping stations, § 322.

Data :—

Liquid : oil of S.G. 0·89 and kinematic viscosity 0·21 stokes.
Pipe : 12-in diameter (assumed horizontal).
Discharge : 1200 gall./min.
Permissible maximum pressure in pipe : 600 lb./sq. in.

Method :—

(i) Velocity in pipe = 4·1 ft./sec.

Kinematic viscosity in foot units = 0·00023 sq. ft./sec.

$$\text{Reynolds number in pipe} = \frac{vd}{\nu} = \frac{4{\cdot}1 \times 1}{0{\cdot}00023} = 17{,}800.$$

If the pipe were smooth, the corresponding value of the pipe coefficient "f" would be about 0·0066. For the steel pipe actually used, the coefficient would probably be about 0·0085.

$$\text{Head loss between stations} = \frac{600 \times 2{\cdot}31}{0{\cdot}89} = 1560 \text{ ft.}$$

$$= \frac{4 \times 0{\cdot}0085 \times L}{1{\cdot}0} \times \frac{(4{\cdot}1)^2}{64{\cdot}4}.$$

From which L = distance between stations = 175,000 ft = about 34 miles.

(ii) $$\text{Output power at each station} = \frac{1200 \times 8{\cdot}9 \times 1560}{33{,}000} = 505.$$

The duty might thus be divided between two pumping units, with a third as standby.

Allowing a pump speed of 3000 r.p.m., and a specific speed per stage of 20 (foot), we find that to correspond with the stipulated discharge of 1200 gall./min. the *head per stage* would be 256 ft.

Therefore number of stages per pump $= \frac{1560}{256} = 6{\cdot}1$ stages.

A 6-stage pump could therefore be arranged, of the type shown in Fig. 80.

(iii) Because of the effect of the oil viscosity on pump performance, it would be prudent to allow a pump efficiency of about 0·65, necessitating an input power of

$$\frac{505}{0{\cdot}65} \times \tfrac{1}{2} = 390 \text{ h.p. per unit.}$$

Probably engines running at 600 r.p.m. might be recommended, driving the pumps through 5 : 1 ratio step-up gears.

(iv) The provisional figures just given should not be accepted without thorough discussion. Although the first cost of the installation might be reasonably low, the pumping costs could easily mount up very considerably. Possibly a slight increase in pipe diameter would bring about a more than proportionate reduction in the number of pumping stations. In any event, centrifugal pumps might not offer the best solution at all. Slow-speed reciprocating pumps would certainly be more efficient and durable.

EXAMPLE 44.

To plan a boosting installation, § 324.

Data :—

Diameter of water main = 36 in.
Length of water main = 34 miles.
Gravitational head = 198 ft.
Existing flow = 12 millions of gallons per day.
Boosted flow, after boosting installation has been installed in main = 15 millions of gallons per day.

Method :—

(i) In such an important water main as this, it is improbable that boosting will be considered at all until the main has been in service for a number of years. The pipe surface can therefore be taken to be "rough", hence the frictional loss will vary as the *square* of the discharge.

Thus, new frictional loss $= 198 \times \left(\frac{15}{12}\right)^2 = 310$ ft. Of this amount, the booster pumps must supply $310 - 198 = 112$ ft.

(ii) The maximum flow of 15 millions of gallons per day corresponds to 10,400 gall./min , and the maximum power output of the boosting station is therefore

$$\frac{10{,}400 \times 10 \times 112}{33{,}000} = 350 \text{ h.p.}$$

The system of Fig. 216 appears suitable, using two pumps at maximum flow, and three pumps at low discharges. It might be considered possible to dispense with any further plant, because if necessary two pumps always *could* do the work, leaving the third as a standby.

(iii) The duty of each pump can now be specified, thus :—

Head = 112 ft.
Discharge = 5200 gall./min.

Standard types of double-inlet centrifugal pumps would be acceptable, of specific speed about 30 (foot). Their maximum speed would be about 820 r.p.m., permitting them to be direct-coupled to variable-speed direct-current motors. A pump efficiency of at least 85 per cent. should be realised, and the motor b.h.p. need therefore not exceed 210 per pump.

(iv) Here again the boosting system cannot be taken as the sole solution of this particular problem of supplying more water. Another water main, laid alongside the first, would do all that is required. A study of predicted operating conditions and detailed comparative costs, extending over a long term of years, will alone provide the correct answer.

EXAMPLE 45.

To study the effect of outlet conditions on a pumping plant, § 328.

Data :—

The plants to be compared are those shown in Fig. 221.
Constant discharge = 17 cu. ft /sec. per pump.
Maximum static lift = 12 ft.
Minimum static lift = 6 ft.
Pipe diameter = 15 in.; outlet 2 ft. above highest water surface.
Enlarged or flared outlet diameter = 24 in.

Method :—

As the pipe velocity works out at 13·8 ft./sec., the pipe friction will amount to about 1 ft., and the velocity head to about 3 ft.

But with the flared outlet the velocity head wasted will only be about 0·5 ft.

The pump efficiency may be taken as 0·8, except for low-head conditions in system (ii), when it might fall to 0·7.

In the following tabular statement, P_{st} represents the net power corresponding to the static head, P_s represents the power input to the pump, and therefore η_{st} is the static efficiency $= P_{st}/P_s$, § 165 (iii).

System.	H_s ft	H_e ft	Q cu ft /s.	P_{st}.	P_s.	η_{st}.
No bellmouth, No siphonic effect (i)	12	18	17	23·1	43·7	0·53
	6	18	17	11·6	43·7	0·27
Bellmouth outlet with siphonic effect (ii)	12	14	17	23·1	33·9	0·68
	6	8	17	11·6	22·1	0·52

The figures show clearly enough the advantage of submerging the outlet pipe, if this is practicable on other grounds.

EXAMPLE 46.

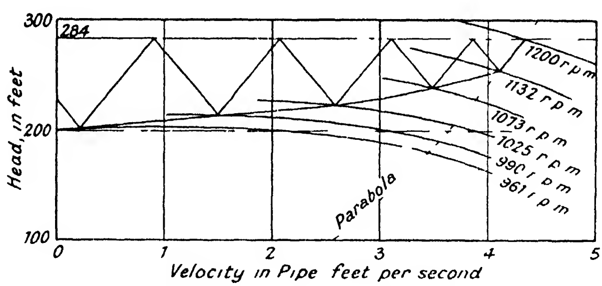
In an electrically-driven pumping set, to show the effect of a fly-wheel in reducing surge pressures, § 334.

Data :—

As in Example 35, except that now each pump has a fly-wheel mounted on its shaft, having a moment of inertia of 5500 lb ft.^{2} The total moment of inertia of each set of revolving parts is therefore $1450 + 5500 = 6950$ lb.ft.^{2}

Method :—

As in Example 35, except that as the pump speed falls more gradually than before, it is permissible to compute speed changes



corresponding to periods $2dt$ instead of dt. The graphical method (see diagram) gives the following results, which may be compared with those computed in Example 35.

Minimum head at pumps = 200 ft.
Time for flow through pumps to cease = 2·4 sec.
Negative surge = 284 − 200 = 84 ft.
Maximum head, resulting from positive surge = 368 ft.
Pump speed at moment when flow through pump ceases = 950 r.p.m.

Although, then, the beneficial action of the fly-wheels is very marked, it is not proportional to the added weight. In order to

reduce the surge pressure by one-half, the moment of inertia of the revolving parts has been increased more than four-fold.

EXAMPLE 47.

To compare propeller and centrifugal pumps for supplying circulating-water to surface condensers, § 342.

Data :—

Each pump is to supply 22,000 gall./min. against a total head of 24 ft. The diameter of the pipes is 32 in.

Method :—

(i) *Propeller pump.* It would make a neat arrangement if the rotor diameter were equal to the pipe diameter. Moreover, in order to give the pump a reasonable suction performance, a two-stage system may be studied. In a general way, the pump would be mounted horizontally, somewhat as shown in Fig. 223 (i).
To suit the stipulated conditions :—

Head per rotor = 12 ft.
Rotor diameter = 2·67 ft.
Speed ratio = 2·2,

it will be found that : Speed $N = 440$ r.p.m Specific speed $N_s = 178$ (foot), both of which may provisionally be accepted. For other systems of intake conduits, a vertical single-stage arrangement on the lines of Fig. 68 may be studied. Discharge variation, which permits the power consumption of the pumps to be reduced in cold weather, can be effected either by the use of variable-speed motors or by using variable-pitch propellers, § 104.

(ii) *Centrifugal pumps.* A specific speed of about 75 (foot) is usually found satisfactory. If a single impeller had to deal with the full discharge of 22,000 gall./min. against the full head of 24 ft., it would have to run at 320 r.p.m. But if twin pumps working in parallel were used, as in Fig. 212 (*a*), then the speed could be raised to 450 r.p m.
As none of the specific speeds provisionally chosen are close to limiting values, the final value adopted can readily be adjusted, if necessary, to suit the nearest permissible speed of the driving motor.

EXAMPLE 48.

To estimate the maximum permissible water temperature in motor-driven and in turbine-driven boiler feed pumps.

Data :—

Each pump is required to discharge 676 gall./min. against a total head of 1083 ft.

The motor-driven pumps have eight stages and run at 1475 r.p.m.

The turbo-driven pumps have two stages and run at 4000 r.p.m. The condensate-extraction pumps generate a head of 190 ft.

Method :—

(i) *Motor-driven pumps*. The head per stage is 135 ft., and the usual formula shows that the specific speed per stage is 17 (foot). In turn the cavitation factor σ for a single-inlet impeller is found to be 0·18, § 253, whence $H_{int} = \sigma \times H_{es} = 24$ ft. This corresponds to a pressure difference of 10 lb./sq. in.
Now the head generated by the extraction pump—190 ft. —represents a pressure difference of 82 lb./sq. in. For present purposes (neglecting the absolute pressure in the condenser), this means that at the extraction pump delivery flange the absolute pressure is 82 lb./sq. in. Deducting from this amount the pressure loss between extraction pump and feed pump inlet—which can be taken at the high value of 30 lb./sq. in. so as to keep a margin in hand—it follows that the absolute pressure at feed pump inlet is $82 - 30 = 52$ lb./sq. in.
On applying the formula of § 345 (v), it appears that

$$52 = p_{vp} + wH_{int} = p_{vp} + 10,$$

or p_{vp} = vapour pressure of water in feed pump = 42 lb./sq. in.
The corresponding temperature is 270° F., which is the maximum temperature to which the feed-water could be heated before entering the feed-pump.

(ii) *Turbo-driven feed-pump*. For these pumps the corresponding values are :—

$$H_{es} = \text{head per stage} = 542 \text{ ft.}$$
$$N_s = \text{specific speed} = 16 \text{ (foot)},$$
$$\sigma = \text{cavitation factor} = 0{\cdot}16,$$
$$H_{int} = 0{\cdot}16\, H_{es} = 87 \text{ ft.}$$

Thus $52 = p_{vp} + (w \times 87)$, or $p_{vp} = 16$ lb./sq. in.
The limiting feed-water temperature is therefore 216° F.

(iii) Although, therefore, the motor-driven pumps could work at a higher temperature than the turbo-pumps, in practice it would not be possible to differentiate between them, because either type of pump may be called upon for duty at any moment. The feed-heating arrangements would therefore allow for a water temperature at feed-pump inlet of not more than 216° F. Two factors of safety have already controlled the computations: the maximum value of "sigma", and the high value of the pressure loss in the system, viz. 30 lb./sq. in. In comparison with these, only an insignificant error is involved in using a nominal value for the density w_{tf} instead of the true one.

KEY TO SYMBOLS GENERALLY USED IN THE BOOK

(Special notations are explained in individual paragraphs)

A, a	area.
b	breadth or width.
B.H.P.	brake horse-power.
c	chord length of aerofoil or blade element.
C	constant or co-efficient.
C_D	drag coefficient of aerofoil or blade element.
C_L	lift coefficient of aerofoil or blade element.
D, d	diameter.
D	drag on aerofoil or blade element.
E	energy per unit weight of liquid.
E_∞	energy per unit weight of liquid, imparted by ideal rotor having an infinite number of blades.
E_n	energy per unit weight of liquid, imparted by actual rotor having n blades.
f	pipe coefficient for evaluating friction loss in closed passages.
g	acceleration of gravity.
H, h	pressure-head: energy per unit weight: difference of surface level.
h_a	head equivalent to atmospheric pressure.
h_c	characteristic head number.
h_d	differential pressure-head between back and front of rotor blade.
h_{dd}	dynamic depression head in rotor passages.
H_e	effective head: effective energy per unit weight given to liquid while flowing through pump.
h_f	frictional loss of head in pipe or closed passage.
h_g	gain of pressure-head in recuperator.
h_i	pressure-head generated in ideal impeller.
h_i	inertia head in pipes during starting or stopping of pump.
h_l	eddy or turbulence loss in pipe or closed passage.
H_m	total manometric head generated by pump.
H_{md}	manometric delivery head.
H_{ms}	manometric suction head.
H_s	total static lift or dead head.
h_s	static suction lift.
h_v	velocity energy or head.
h_{vp}	head of liquid equivalent to vapour pressure of liquid.
K, k	a constant or coefficient.
K_p, k_p	horse-power constant: energy per second equivalent to one horse-power.
K_s	specific speed constant.
L, l	length.
L	lift on aerofoil or blade element.
n	number of blades in rotor: speed in revolutions per second.

N	rotational speed in revolutions per minute.
n_s	characteristic shape number of rotor or pump.
N_s	true specific speed of rotor or pump.
N_{sn}	nominal specific speed.
P	force or thrust: power: horse-power.
P_d	dynamic thrust.
P_s	shaft horse-power input to pump.
P_w	"water" horse-power output of pump.
p	pressure: intensity of pressure.
p_a	atmospheric pressure.
p_{vp}	vapour pressure of liquid.
Q, q	rate of discharge: volume flowing in unit time.
q_c	characteristic discharge number.
q_l	volumetric leakage or slip loss.
R, r	radius.
r.p.m.	revolutions per minute.
R.V.	reflux valve.
S.G.	specific gravity.
S.H.P.	shaft horse-power input to pump.
T, t	thickness: torque.
U	absolute velocity of liquid.
v	peripheral velocity of a point on a revolving blade or rotor: mean velocity in a pipe or passage.
v_r	relative velocity of liquid flowing through a rotor passage.
V	tangential or whirl velocity component of liquid.
W	weight: weight of liquid flowing in unit time.
w	density of liquid: weight per unit volume.
W.H.P.	water horse-power output of pump.
Y	velocity of flow (in a radial-flow pump, radial component of velocity).
Y_r	in general, radial velocity component of liquid.
Y_a	axial velocity component.
Y_m	meridional velocity component.

α	Coriolis acceleration, i.e., tangential acceleration of liquid: angle of attack of aerofoil or blade element.
β	inlet blade angle of rotor.
γ	outlet blade angle of rotor.
δ	inlet blade angle of recuperator.
Δ	relative power loss (with distinguishing subscript).
ϵ	relative blade loading.
η_m	gross or overall efficiency of pump.
η_h	hydraulic efficiency of pump.
λ	width ratio of rotor.
ν	kinematic viscosity.
σ	Thoma cavitation factor.
ϕ	speed ratio of rotor.
ψ	flow ratio of rotor.
ω	angular velocity.

KEY TO SYMBOLS

The following *subscripts* denote the part of the pump or rotor to which a particular symbol refers :—

0 inlet to pump casing.
1 inlet to rotor.
2 outlet from rotor.
3 outlet from pump casing.

In an axial-flow rotor :—

a inner radius.
b outer radius.

Thus, Y_2 denotes the velocity of flow at the outlet from a centrifugal pump rotor.

Other subscripts are :—

n relating to an actual rotor having *n* blades.
∞ relating to an ideal rotor having an infinite number of blades.

TABLE OF CONVERSION FACTORS

Usually-accepted values are here given. The asterisk (*) denotes that the factor depends upon temperature or other variable, and that the figure quoted is based upon normal atmospheric temperature—say 50° F. Unless otherwise stated, *gallon* means Imperial gallon = 0·1605 cubic feet.

	To Convert	To	Multiply by
Weight	Pounds	Grams	453·6
	Pounds	Kilograms	0·4536
	Kilograms	Pounds	2·205
	Tons	Kilograms	1016
	Tons (2240 lbs.)	Metric tons (1000 kg.)	1·016
Length	Inches	Centimetres	2·540
	Feet	Centimetres	30·48
	Feet	Metres	0·3048
	Metres	Feet	3·281
	Yards	Metres	0·9144
	Miles	Kilometres	1·6093
Speed or Velocity	Feet per second	Centimetres per second	30·48
	Feet per second	Metres per second	0·3048
Area	Square inches	Square centimetres	6·452
	Square feet	Square centimetres	929·0
	Square feet	Square metres	0·0929
	Square metres	Square feet	10·76
	Square yards	Square metres	0·8361
	Acres	Hectares	0·4047
Pressure and Head	Pounds per square inch	Kilograms per square centimetre	0·0703
	Kilograms per square centimetre	Pounds per square inch	14·22
	Grams per square centimetre	Dynes per square centimetre	981 *
	Pounds per square inch	Feet head of water	2·31*
	Feet head of water	Pounds per square inch	0·434*
	Pounds per square inch	Inches head of mercury	2·04*
	Pounds per square inch	Atmospheres	0·0681*
	Kilograms per square centimetre	Metres head of water	10·00*
	Inches head of mercury	Millibars	33·87*
	Millibars	Dynes per square centimetre	1000·00

TABLE OF CONVERSION FACTORS

	To Convert	To	Multiply by
Volume or Capacity	Cubic inches	Cubic centimetres	16·39
	Cubic feet	Cubic centimetres	28320
	Cubic feet	Cubic metres	0·02832
	Cubic metres	Cubic feet	35·31
	Cubic yards	Cubic metres	0·7645
	Cubic feet	Litres	28·32
	Cubic feet	Imperial gallons	6·23
	Imperial gallons	Cubic feet	0·1605
	Imperial gallons	Litres	4·546
	Imperial gallons	American (U.S.) gallons	1·20
	Acre-feet	Cubic feet	43560
Density	Pounds per cubic inch	Grams per cubic centimetre	27·67
	Pounds per cubic foot	Kilograms per litre	0·01602
Viscosity	Foot units $\left(= \frac{\text{lb. sec.}}{\text{sq. ft.}}\right)$	Poises $\left(= \frac{\text{dynes sec.}}{\text{sq. cm.}}\right)$	479
Kinematic viscosity	Foot units (= sq. ft. per sec.)	Stokes (= sq. cm. per sec.)	929
Temperature	Degrees Fahrenheit	Degrees Centigrade	$\frac{°\text{F.} - 32}{°\text{F.}} \times \frac{5}{9}$
Rate of Discharge	Cubic feet per second	Gallons per minute	374
	Cubic feet per second	Millions of gallons per 24 hours	0·538
	Cubic feet per second	Tons of water per minute	1·67*
	Cubic feet per second	Litres per second	28·32
	Cubic feet per second	Cubic metres per second	0·0283
	Gallons per minute	Litres per second	0·0757
Energy: Power	Foot-pounds	Kilogram-metres	0·1382
	Horse-power (550 ft. lb./sec.)	Metric horse-power (75 kg. m./sec.)	1·014
	Horse-power	Kilowatts	0·746
	Metric horse-power	Kilowatts	0·736

PHYSICAL PROPERTIES OF WATER AND OF SATURATED WATER VAPOUR

Temperature.		Density of Water at Saturation Pressure.		Vapour Pressure.	
Deg. Fahr.	Deg. Cent.	Lb. per cubic foot.	Kilograms per litre.	Lb. per square inch.	Equivalent head of water at given Temperature. (feet.)
39	4	62·4	1·000	0·12	0·27
50	10	62·4	1·000	0·18	0·41
68	20	62·3	0·998	0·34	0·78
86	30	62·2	0·996	0·61	1·4
104	40	62·0	0·992	1·1	2·5
140	60	61·4	0·983	2·8	6·6
176	80	60·6	0·972	6·9	16·4
212	100	59·8	0·959	14·7	35·6
248	120	58·9	0·944	28·5	69·8
284	140	57·9	0·927	52·2	130
320	160	56·8	0·909	89·9	229
356	180	55·6	0·889	144	376
392	200	54·1	0·866	226	604
428	220	52·5	0·841	337	924
464	240	50·7	0·814	486	1380

MEAN PRESSURE OF THE ATMOSPHERE

Altitude above Sea-level.	feet.	0	2000	4000	6000	8000
	metres.	0	610	1220	1830	2440
Barometric pressure in lb. per sq. inch . .		14·7	13·7	12·7	11·8	10·9
Barometric Pressure as a Percentage of pressure at sea-level . . .		100	93	86	80	74

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Jour. Inst. C.E. = Journal of the Institution of Civil Engineers.
Proc. I. Mech. E. = Proceedings of the Institution of Mechanical Engineers.
Trans. A.S.M.E. = Transactions of the American Society of Mechanical Engineers.
Engg. = *Engineering* (London).
The Eng. = *The Engineer* (London).
Pwr. = *Power* (New York).
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INDEX

Unless otherwise stated, the numbers refer to *Paragraphs* (§§), and the *letters* refer to the *Part* of the book or the nature of the information. (A = Principles, B = Design or Construction, C = Performance, D = Installation.) Thus, to find information about the *design* of axial-flow pumps, the Index shows that paragraphs 103, 104, 105, 109-113 should be consulted. Paragraph reference numbers are printed at the *top* of the pages.

References to the Illustrative Examples are always given in the Text, but those likely to be of special importance are mentioned also in the Index.
